MATHEMATICS

NATURAL PHILOSOPHY

MECHANICAL THEORY

MACHINE CONSTRUCTION

SCIENCE AND ART

CHEMISTRY

Collins' Advanced Science Series.

PRINCIPLES

OF

MACHINE CONSTRUCTION;

BEING AN APPLICATION OF

GEOMETRICAL DRAWING

FOR

THE REPRESENTATION OF MACHINERY

BY

EDWARD TOMKINS,

AUTHOR OF "MACHINE CONSTRUCTION AND DRAWING."

EDITED BY

HENRY EVERS, LL.D.,

AUTHOR OF "STEAM AND THE STEAM ENGINE," "NAVIGATION," ETC. ETC.

VOL. I.—TEXT.

LONDON AND GLASGOW:

WILLIAM COLLINS, SONS, AND COMPANY

1878.

PREFACE.

At the request of the Publishers, Mr. Edward Tomkins, who had written on the same subject in their Elementary Series, undertook the preparation of this work, and had been engaged upon it for a considerable time, when he was interrupted by ill health. This illness, which eventually rendered necessary a dangerous surgical operation, did not prevent him from giving what strength he had left to the completion of the work he had taken in hand. To the regret of all who knew him, however, and in spite of all that care and skill could do, his life was suddenly cut short, just as this work was nearly finished.

The MSS. and Drawings were then placed in the hands of the present Editor, who found the plan of the work so clearly and distinctly laid down, that to follow it was comparatively easy.

His task has been to see the work through the press, to endeavour to supply details which seemed to have been left for further search or information, and so far as he could to correct errors, and verify references, and at the same time add to the articles, plates, and figures, necessary for the completion of the original plan.

Those who have had to deal with the unfinished MSS. of others can understand some of the difficulties of the task

imposed on the Editor, while from those who have been fortunate enough not to be so situated the Editor claims indulgence, as he has used his utmost ability to minutely carry out Mr. Tomkins' plan, not only as a matter of honesty, but because the more he studied it, he conscientiously believed it to be the best for the purpose for which the book was designed.

Accompanying the work are a number of Plates; it is hoped that these will not only illustrate the various points discussed in the body of the text, but supply good Drawing Copies for advanced Students in Science and Art Classes.

Teachers using the diagrams for class instruction, are earnestly advised not to employ them as mere copies, but to give either "plan" or "elevation" on the black-board, and then request from the student a drawing of the elevation or plan, or else to give a general description of the object, accompanied by a rough sketch and proper dimensions, and then require a worked drawing of the object chosen.

Notwithstanding the peculiarly adverse circumstances under which this work has been produced, the Editor and Publishers, knowing the marked ability of the Author, and the success of his Elementary work, confidently commend the present volume to the public.

March, 1878.

CONTENTS.

CHAPTER I.

CHAPTER II.

CHAPTER III.

CHAPTER IV.

CHAPTER V.

CHAPTER VI.

CHAPTER VII.

On the Transmission of Motion by Wheel-work.

CHAPTER VIII.

ON THE TEETH OF WHEELS.

CHAPTER IX.

TABLES.

MACHINE CONSTRUCTION AND DRAWING.

CHAPTER I.

SECTION I.

DRAWING APPARATUS.

1. Drawing Apparatus.—The student should possess a drawing-board, a T-square, two set-squares, a box of instruments, drawing paper, a rule, scales, india-rubber, drawing pins, and pencils, for machine construction and drawing.

The size of the *drawing-board* will depend upon the kind of work which the student has to execute; the first of the following sizes will be sufficient for ordinary private practice, but for drawing-office purposes the largest size will be required, and often a still larger size :—2 ft. × 1 ft. 6 in.; 2 ft. 4 in. × 1 ft. 9 in.; 3 ft. 6 in. × 2 ft. 6 in.

The T-*square* should have a blade not shorter than the long edges of the drawing-board, see fig. 8, and for general use the stock and blade should be permanently fixed together. In some cases it is convenient to have a T-square with a movable stock, so that lines not parallel to the edges of the board may be drawn by means of it.

The *set-squares* should have the angles shown in figs. 1 and 2; the one represented in fig. 1 should have its shortest edge not less than 4 in., and that in fig. 2 its short edges not less than 5 in.

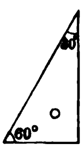


Fig. 1.

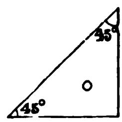


Fig. 2.

2. The *box of instruments* should contain at least the following :—A pair of dividers, a set of large compasses with lengthening bar and with pen and pencil legs, small pen and pencil bows, a drawing pen, and a protractor. In addition

to these there may be added with advantage a smaller set of compasses with pen and pencil legs, a small sized drawing pen, small spring pen and pencil bows, spring or hair dividers, and proportional compasses; the last are extremely useful for making drawings of an enlarged or reduced size. In addition to the instruments named there are others in use, but as they are of a special character and only used in certain cases, we shall not enumerate them here. Instruments with needle points are the best, but the needle should have a shoulder or collar upon it to prevent it making too large a hole in the paper.

3. The *drawing paper*, used both as regards size and quality, must depend upon circumstances; for ordinary practice *cartridge* drawing paper may be used, it can be obtained in sheets of various sizes and in rolls. The kind of paper generally used for office work and shaded drawings is *double elephant*, of which there are two kinds, smooth and rough; the former is the best for fine-lining and shading The following table gives the names and sizes of the sheets of drawing paper in general use; the sizes may be found to vary a little :—

TABLE I.

SIZES OF DRAWING PAPER.

Antiquarian, ...	4 ft. 4½ in. × 2 ft. 6½ in.
Double Elephant, ...	3 ,, 4 ,, × 2 ,, 2½ ,,
Atlas, ...	2 ,, 9½ ,, × 2 ,, 2 ,,
Imperial, ...	2 ,, 6 ,, × 1 ,, 9 ,,
Elephant, ...	2 ,, 4 ,, × 1 ,, 10½ ,,
Royal, ...	2 ,, 0 ,, × 1 ,, 7½ ,,
Medium, ...	1 ,, 10 ,, × 1 ,, 5½ ,,
Demy, ...	1 ,, 8 ,, × 1 ,, 3½ ,,

Cartridge can be obtained in sheets 4 ft. 6 in × 2 ft. 3 in., and in rolls 4 ft. 9 in. or 4 ft. 4 in wide.

4. The *drawing pencils* to be used must depend upon the kind of drawing the student is engaged upon; the best for ordinary use are those marked H, for fine work HH, and for very fine work HHH. For sketching and ordinary freehand work those marked F or HB are to be used. The pencils to be used with the squares should have a flat chisel-point, as shown in figs. 3 and 4; the broad side to be kept in contact

with the square, while those for sketching, etc., should have the ordinary conical or round point. The flat chisel-point will last a considerable time compared with the ordinary round point; we advise the beginner to be very careful in sharpening his pencils. After cutting the pencil with a knife it may be finished for use by rubbing it upon a smooth file or a piece fine sandpaper; and a finer point still may be got by rubbing it upon a piece of drawing paper.

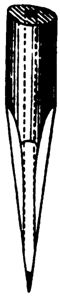


Fig. 3.

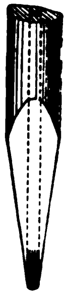


Fig. 4.

5. *Drawing pins* may be used for fastening the paper to the board if the drawing is not very elaborate, or if it is to remain upon the board for a few days only; but if it is to be coloured or shaded, then it should be stretched and fixed with glue or gum; see articles on working and finished drawings.

Fine vulcanised *india-rubber* is the best for ordinary use, but if the drawing is to be coloured the common dark kind should be used. In all cases, the india-rubber should be used as little as possible, since by too frequent rubbing the paper becomes rough and more liable to get dirty.

SECTION II.

STANDARDS OF MEASURE.

6. British Standard.—The dimensions of a machine are expressed in terms of a certain standard. In Great Britain the *foot* is the standard, the one-third of the *standard yard*, which is the distance at the temperature of 62° Fahrenheit, between two marks on a certain bar of platinum kept in the office of the exchequer. The standard foot is divided into twelve equal parts, called *inches;* these are subdivided into 2, 4, 8, 16, 32, etc., equal parts, and termed respectively *halves, quarters, eighths, sixteenths, thirty-seconds*, etc., denoted

by $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{16}$, $\frac{1}{32}$, etc., each new subdivision being one-half of the preceding.

The instrument used for taking dimensions is termed a *rule*, and the common "two-foot" is the one in general use. The "two-foot rule" consists of two pieces of wood or metal, each one foot long, which are jointed together for convenience; the ordinary form is represented in fig. 5, which shows a small portion of the rule with the above subdivisions.

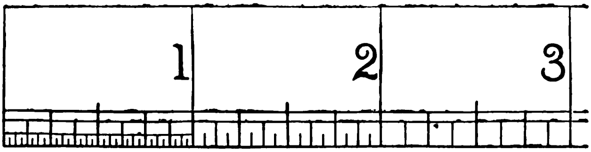


Fig. 5.

7. The *decimal rule* is used in some engineering establishments, and will in all probability come into general use before many years are past, as it possesses advantages which the one now more commonly used does not; we therefore describe it, and add, for convenience of reference, a couple of tables of *decimal equivalents* of inches and fractions of an inch; see Tables II. and III.

In the decimal rule the inch is taken as the standard, and there are ten inches in each half of the rule, instead of twelve, as in the "two-foot;" each inch is divided into ten equal parts, these are again subdivided into hundredths, thousandths, etc.

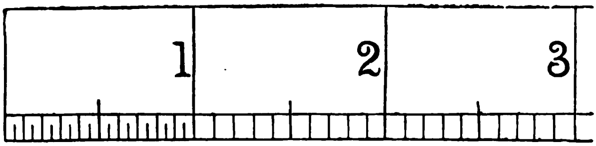


Fig. 6.

Fig. 6 represents a portion of the decimal rule, upon which is shown inches and tenths of an inch, the latter are divided into two equal parts representing ·5 of a tenth of an inch, or ·*05 of* an inch.

TABLE II.

DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH.
ONE INCH TAKEN AS UNITY.

Fractions of an Inch.	Decimals of an Inch.	Fractions of an Inch.	Decimals of an Inch.	Fractions of an Inch.	Decimals of an Inch.	Fractions of an Inch.	Decimals of an Inch.
$\frac{1}{64}$ =	·01562	$\frac{1}{4}$ =	·25	$\frac{1}{2}$ =	·5	$\frac{3}{4}$ =	·75
$\frac{1}{32}$ =	·03125	$\frac{9}{32}$ =	·28125	$\frac{17}{32}$ =	·53125	$\frac{25}{32}$ =	·78125
$\frac{1}{16}$ =	·0625	$\frac{5}{16}$ =	·3125	$\frac{9}{16}$ =	·5625	$\frac{13}{16}$ =	·8125
$\frac{3}{32}$ =	·09375	$\frac{11}{32}$ =	·34375	$\frac{19}{32}$ =	·59375	$\frac{27}{32}$ =	·84375
$\frac{1}{8}$ =	·125	$\frac{3}{8}$ =	·375	$\frac{5}{8}$ =	·625	$\frac{7}{8}$ =	·875
$\frac{5}{32}$ =	·15625	$\frac{13}{32}$ =	·40625	$\frac{21}{32}$ =	·65625	$\frac{29}{32}$ =	·90625
$\frac{3}{16}$ =	·1875	$\frac{7}{16}$ =	·4375	$\frac{11}{16}$ =	·6875	$\frac{15}{16}$ =	·9375
$\frac{7}{32}$ =	·21875	$\frac{15}{32}$ =	·46875	$\frac{23}{32}$ =	·71875	$\frac{31}{32}$ =	·96875

TABLE III.

DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH AND OF A FOOT.
ONE FOOT TAKEN AS UNITY.

Fractions of an Inch.	Decimals of a Foot.	Fractions of an Inch.	Decimals of a Foot.	Inches.	Decimals of a Foot.	Inches.	Decimals of a Foot.
$\frac{1}{32}$ =	·002604	$\frac{1}{2}$ =	·0416$\dot{6}$	2 =	·166$\dot{6}$	7 =	·583$\dot{3}$
$\frac{1}{16}$ =	·005208	$\frac{5}{8}$ =	·05208	3 =	·25	8 =	·666$\dot{6}$
$\frac{1}{8}$ =	·01041$\dot{6}$	$\frac{3}{4}$ =	·0625	4 =	·333$\dot{3}$	9 =	·75
$\frac{1}{4}$ =	·02083	$\frac{7}{8}$ =	·07291$\dot{6}$	5 =	·416$\dot{6}$	10 =	·833$\dot{3}$
$\frac{3}{8}$ =	·03125	1 =	·08333	6 =	·5	11 =	·916$\dot{6}$

8. The *measure of area* is the *square inch;* sometimes the square foot is employed, which contains 144 square inches.

The *measure of volume*, or solid measure, is the *cubic inch;* sometimes the cubic foot is employed, which contains 1728 cubic inches.

The *measure of weight* is the *avoirdupois pound*, which contains 16 *ounces* = 256 *drams* = 7000 *grains;* sometimes the *ton* is employed, which contains 2240 such pounds.

9. French Standard.—In France, Belgium, and some other Continental countries, the *metre* is the standard measure of length; the metre is equal to 39·3704 British inches, and is divided into decimetres, centimetres, and millimetres. The

following are the Britith equivalent values of the divisions of the metre expressed in decimals :—

		Of the Metre.	In British Inches.
French.	Millimetre,	0·001	0·0393
	Centimetre,	0·01	0·3937
	Decimetre,	0·1	3·9370
	Metre,	1·	39·3704

The *measures of area* are the *square millimetre, centimetre, decimetre,* and *metre.* The square metre contains 100 square decimetres = 10,000 square centimetres = 1,000,000 square millimetres.

The *measures of volume* are the *cubic millimetre, centimetre, decimetre,* and *metre.* The cubic metre contains 1000 cubic decimetres = 1,000,000 cubic centimetres = 1,000,000,000 cubic millimetres.

The *measure of weight* is the *kilogramme,* which is equal to 2·20462125 avoirdupois pounds.

SECTION III.

CIRCUMFERENCE OF A CIRCLE—NOTATION AND SIGNS.

10. Circumference of a Circle.—The circumference of a circle is often required for calculations and for drawing purposes, we therefore give examples showing how it should be found.

I. *By calculation.*—The circumference of a circle expressed in terms of its *diameter* is an incommensurable ratio usually denoted by the Greek letter π. The approximate value of this ratio generally used is 3·1416; that is, the circumference is taken as 3·1416 times the diameter. Let D = the diameter, then the circumference = π D; where π is the constant, and D the variable.

Example.—Let D = 3·5″; find the circumference.

$$\begin{aligned} \text{Circum.} &= \pi D \\ &= 3{\cdot}1416 \times 3{\cdot}5'' \\ &= 10{\cdot}9956 \text{ inches.} \end{aligned}$$

The following is a list of other approximations used, together with their errors; the values are in each case in excess of the true value:—

Value of π	Error (about)	Value of π	Error (about)
3·141593	$\frac{1}{9,000,000}$	$\frac{355}{113}$	$\frac{1}{13,000,000}$
3·1416	$\frac{1}{400,000}$	$\frac{22}{7}$	$\frac{1}{2500}$

II. *By Chords.*—In some of the constructions in this book we have given an approximate, as well as a correct, method of setting out the circumference of a circle. The approximate method is that of substituting for an *arc* the corresponding *chord* of the circle; if the arc taken is small, the error introduced by this substitution will also be small. The circumference of a circle expressed in terms of its *radius* is denoted by 2π, where π equals the ratio before stated.

Let r = the radius, then
$2\pi r$ = the circumference.

In fig. 7, ABD is a circle, AC and BC are radii; AB is the chord subtending the angle ACB, and AFB is the corresponding arc. The angle ACB is bisected by CF, and therefore the chord AB is bisected in E. The ratio $\frac{AE}{AC}$ is the *sine* of the angle ACE; the chord AB is equal to twice the sine of half the angle ACB. By means of a table of sines the value of AE, and therefore of AB, can be found for any given angle.

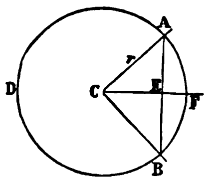


Fig. 7.

11. The following table gives the circumference as determined by chords of different length:—

TABLE IV.

CIRCUMFERENCE OF A CIRCLE BY CHORDS.

No. of Chords.	(1) Value of 2π by Calculation.	(2) Value of 2π by Chords.	(3) Difference, or (1) — (2).	(4) Error.
48	6·2832	6·2784	0·0048	$\frac{1}{1309}$
24	6·2832	6·2648	0·0184	$\frac{1}{341}$
12	6·2832	6·2116	0·0716	$\frac{1}{88}$
6	6·2832	6·0	0·2832	$\frac{1}{22}$

The foregoing table gives also the difference and the error existing between the results obtained by the method of calculating by arcs and that by chords; the radius r is taken as unity.

12. The use of Table IV. can be best explained by a few examples:—

Examples.—1. Suppose we have a circle of 2 inches diameter and divide it into 24 equal chords; if now we set off one of these chords 24 times along a straight line, the length of that line will be 6·2648 inches, from column (2); the calculated value is 6·2832 inches. In this case column (3) shows the error in the decimal of an inch, which is 0·0184, because the radius is unity; and column (4) gives the error as a fraction of the whole circumference, *viz.*, $\frac{1}{341}$ of 6·2832 inches.

2. If the radius is not unity, the error can be found as follows:—Multiply the number in column (3) corresponding to the number of chords by the radius, and the product is the error in decimals of an inch.

13. Notation and Signs.—The following notation and signs are in common use:—For description of the kind of lines employed, see Art. 18, page 19. For dimensions—one accent over a number, as 2′, denotes feet; and two accents, as 6″ denotes inches; thus 1′ 0¼″, means one foot and a quarter of an inch.

The sign of equality is represented by - - - -	$=$
,, addition, or "plus," by - - - -	$+$
,, subtraction, or "minus," by - - - -	$-$
,, multiplication by - - - - - -	$\times$
The ratio 3·1416 : 1 is represented by - - - -	π

Degrees.—A small circle over a number denotes degrees; thus 45° means 45 degrees.

SECTION IV.

THE DRAWING OF LINES—VARIOUS KINDS OF LINES USED.

14. The Drawing of Lines.—For the present we shall confine our remarks to the drawing of pencil lines; the inking-in of drawings will be considered later on, under the heading of working and finished drawings. However, as

engineering drawings are generally "put in" in ink, the student should read that portion of the subject and practise inking-in as soon as circumstances permit.

The pencil lines should be fine and made with as little pressure as possible, consistent with clearness, because if too much pressure is used those portions of the pencil lines which are not covered with ink will leave impressed marks upon the drawing after the black-lead has been removed, and thus tend to disfigure the drawing.

All lines should be drawn from left to right, and should be made sufficiently long at first so as not to require producing; this is advisable on account of the time saved, and also because the lines will be more accurate, as the lengthening of lines tends to cause a break in their continuity. Of course there is a limit to the practical carrying out of this, otherwise a pencilled drawing would be an unintelligible mass of lines; and it is only by practice the student will be able to keep within the limit, but if he adopt a right method at the commencement with simple objects, he will have little difficulty with more elaborate ones. We will now give a few hints that may be useful to the beginner.

15. *Lines parallel* to the long edges of the drawing board should be drawn with the T-square, and also lines at right angles to these, if longer than an edge of the set-square. Fig. 8 shows the T-square in position for drawing such lines.

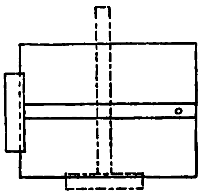


Fig. 8.

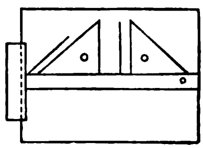


Fig. 9

Fig. 9 shows how the set-squares are used with the T-square for drawing lines at the same angles with the T-square as those of the set-square employed. Parallel lines, which are not parallel to the sides of the board, may be drawn by using the T and set-squares, or by means of two set-squares, as in fig. 10; this method is preferable to that of using a parallel

ruler, as there is less liability to error. Lines at right angles to each other, whatever may be their position with respect to the sides of the drawing board, are best drawn by reversing the set-square after having drawn one of the lines; thus, in fig. 11, *ab* is first drawn, and then the set-square is reversed,

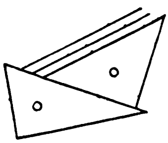


Fig. 10.

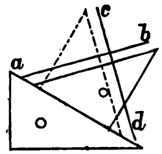


Fig. 11.

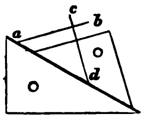


Fig. 12.

as shown in dotted lines, and *cd* drawn, which is at right angles to *ab*. The lines may also be drawn as shown in fig. 12, but they will not be as long, if drawn with the same set-square, as those in the previous example.

16. *Lines inclined* to a given line at an angle which is not one of the angles of the set-squares are to be set off by means of a *protractor*, which is a semicircular piece of horn, brass, or electrum, upon which the degrees are marked; protractors are also made of a rectangular form on six-inch scale rulers, and these are generally the most convenient. The manner of using the protractor is explained thus:—Let AB, fig. 13, be a given line, and C the point in this line through which the required line is to pass; suppose it is required to draw a line inclined downwards from right to left at an angle of 40°. Place the centre of the protractor P upon the point C in AB, and let its diameter coincide with AB; then count from B upwards the required number of degrees, as 40, and make a mark at *a* with a fine pointed pencil. Remove the protractor, and from C through *a* draw the line CD, which is the line making the required angle. If the line is required to slope in the opposite direction, as CE, then the number of degrees must be counted from A upwards.

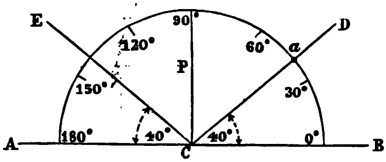


Fig. 13.

Protractors generally have the degrees marked in both directions.

Circles and arcs of circles are to be drawn with the bow pencil, the pencil of which should be cut in a similar manner to those used with the squares, as shown in figs. 3 and 4.

17. *Curved Lines* are often required, for which arcs of circles cannot be substituted, as the curves in Chap. II., Sect. III. and IV., and other curves. To draw these curves wood *moulds, curves*, or *templates*, are used; these moulds are also called French curves. In many drawing offices a number of such curves are kept; but as sometimes only a small portion of a curve is used, and perhaps portions of two curves are necessary to draw the required curve, it is advisable, if a number of similar curves are required, to make a template of the curve by means of which the whole may be drawn. In drawing the threads of the screw in Plates VII. and VIII., we have used such a template. These templates may be made of thin wood, or of cardboard, and may be cut to the required shape with a sharp penknife, and then finished with a little fine sandpaper.

18. **Various kinds of Lines Used.**—The lines of an object which can be seen, as the outline,* are represented in the drawing of that object by full or continuous lines; those lines which cannot be seen, if they are to be represented, are shown by dotted lines, which are of the same thickness as the full lines; for examples of each, see figs. 18 and 19, page 21. This distinction between those lines of an object which can be seen and those which cannot, is always adopted by draughtsmen. In addition to the lines just named there are other distinguishing lines used in this book, which we will now describe. Lines used in determining the form of objects, or *construction lines*, are represented by fine dotted lines. *Centre lines* and *pitch lines*, of which we shall have more to say presently, are represented by alternate long and short dots. Lines representing the radius of a circle, or an arc of one, are termed *radius lines;* these and *dimension lines*, lines on which the dimensions of an object are put, we will now

* By the term *outline* we mean the boundary lines of the object, outside of which there are no lines.

represent. The following is a summary of the kinds of lines used in this book :—

Lines of Objects (Full),	————————————————
,, (Dotted),	....
Construction Lines,	- -
Centre Lines, Pitch Lines,	—— · —— · —— · —— · —— · —— · ——
Radius Lines,	<— - - —— - —— - —— - —— - - —>
Dimension Lines,	<- - - - - - - - - - - - - - - - - ->

The dimension is put in the blank space between the arrow heads in the dimension line. *Shade* or *dark lines* will be introduced and explained under the head of finished drawings.

SECTION V.

DRAWING TO SCALE—SCALES.

19. Drawing to Scale.—Drawings of machinery have generally to be made of a reduced size, on account of the inconvenience and in many cases the impossibility of making them of the same size as the object; such drawings are said to be "drawn to scale." In scale drawings every line in the drawing represents a corresponding line in the object; the *ratio* of the former to the latter represents the *scale of the drawing.*

Let the length of a line in the object $= A$, and let the length of a corresponding line in the drawing $= a$,

$$\text{then the ratio or scale is } \frac{a}{A}$$

For example—Let $A = 4$ inches; $a = 1$ inch, then

$$\frac{a}{A} = \tfrac{1}{4}\text{; the scale is } \tfrac{1}{4}\text{, or 3 inches} = 1 \text{ foot.}$$

I. *Lines.*—In fig. 14 the line a is 2 inches long; suppose it represented by b, which is 1 inch long. The scale would be $\frac{1}{2}$, or 6 inches $= 1$ foot. If a is represented by c, which

is half an inch long, then the scale would be $\frac{1}{4}$, or 3 inches = 1 foot.

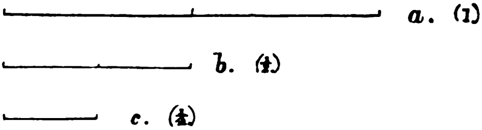


Fig. 14.

II. *Surfaces.*—If instead of a line we have a surface to represent, which of course is bounded by lines, say a square, then the reduction in size is to extend to every line in the drawing. For example—fig. 15 is a square of 1 inch side;

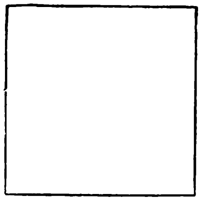


Fig. 15.

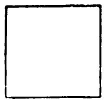


Fig. 16.

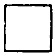


Fig. 17.

a drawing of it to a scale of $\frac{1}{2}$ will be a square of $\frac{1}{2}$ inch side, as fig. 16; one to a scale of $\frac{1}{4}$ will be a square of $\frac{1}{4}$ inch side, as fig. 17.

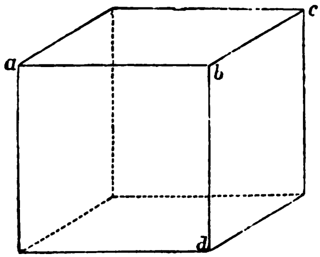


Fig. 18.

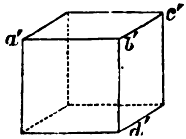


Fig. 19.

III. *Solids.*—The most general case we have to deal with is that of the drawing of solids, in which case, as in the previous one, every line in the drawing must bear a certain

ratio to the corresponding line in the object, and the scale represents this ratio. For example, a cube of 1 inch edge, fig. 18, is required to be drawn to a scale of $\frac{1}{2}$, then every line in the drawing must be made one-half the length of the corresponding line of the object. Fig. 19 is a drawing of the cube to a scale of $\frac{1}{2}$, the lines $a'b'$, $b'c'$, $b'd'$, etc., being respectively one-half the length of ab, bc, bd, etc., in fig. 18.

It must however be observed that the position of the object with respect to the plane of projection* must be considered before this statement is made, as some of the lines in the projection may not bear this ratio; the following statement is more correct:—Suppose we have a full-sized projection of the object, and from it we make a similar projection to a scale of $\frac{1}{2}$, then every line in the latter will be one-half the corresponding line in the former.

20. The scales most frequently used are the following; there are, however, others employed when circumstances do not permit of these being used:—

1, or 12″ = 1 foot.	$\frac{1}{6}$, or 2″ = 1 foot.	$\frac{1}{32}$, or $\frac{3}{8}$″ = 1 foot.
$\frac{3}{4}$, or 9″ = 1 foot.	$\frac{1}{8}$, or $1\frac{1}{2}$″ = 1 foot.	$\frac{1}{48}$, or $\frac{1}{4}$″ = 1 foot.
$\frac{2}{3}$, or 8″ = 1 foot.	$\frac{1}{12}$, or 1″ = 1 foot.	$\frac{1}{64}$, or $\frac{3}{16}$″ = 1 foot.
$\frac{1}{2}$, or 6″ = 1 foot.	$\frac{1}{16}$, or $\frac{3}{4}$″ = 1 foot.	$\frac{1}{96}$, or $\frac{1}{8}$″ = 1 foot.
$\frac{1}{4}$, or 3″ = 1 foot.	$\frac{1}{24}$, or $\frac{1}{2}$″ = 1 foot.	$\frac{1}{192}$, or $\frac{1}{16}$″ = 1 foot.

The reduced dimensions of an object when drawn to scale are taken from an instrument called a "scale," to which we now refer.

21. Scales.—In Art. 19 we have explained the principle of a scale by very simple examples of drawing to scale; we shall now consider the instrument itself.

I. *Plain Scales.*—The ordinary plain scale as used by mechanical engineers is 12 inches long, and is usually made of ivory; boxwood scales are often employed, as they are cheaper as a first cost than ivory ones, though they do not last so long. These scales have marked upon them a number of divided lines, each of which represents a scale; on a scale for ordinary use there should be represented, at least, the following inches and fractions of an inch to the foot:— $\frac{1}{16}$, $\frac{1}{8}$, $\frac{3}{16}$, $\frac{1}{4}$, to 1, advancing by $\frac{1}{8}$ of an inch, $1\frac{1}{4}$ to 2

* See the articles on Projection.

advancing by $\frac{1}{4}$ of an inch, $2\frac{1}{2}$, and 3. One foot of each scale is divided into 12 equal parts representing inches.

II. *Diagonal Scales.*—In addition to the ordinary plain scale there is another form of scale, which is not quite so common, called the diagonal scale, because some of its divisions are made by diagonal lines. The diagonal scale occupies more room than the plain scale, but it is a far more accurate instrument to work with, as will be seen by the examples we shall give of both forms. For examples of scales for drawings, see figs. 20-25, Plate I.

22. Scales are often put upon drawings, and they should always be so, if the scale is small and if the dimensions are not given; whatever contraction takes place in the paper upon which the drawing is, the scale must be contracted in a similar manner. If all the projections of an object on one sheet of paper are drawn to the same scale, we only require one scale on the drawing; if otherwise, more than one is required; we will now give examples of each kind of scale.

23. Fig. 20, Plate I., represents an ordinary plain scale of $\frac{1}{4}$, or 3 inches = 1 foot, showing feet, inches, and quarters of an inch; the left-hand six inches show eighths of an inch. The scale should be numbered as shown, so that if we take with the dividers a certain length of line from the drawing, and apply the dividers to the scale, the length of the line on the object itself can be read off from the scale at once; suppose the length is $1' \, 6\frac{1}{2}''$; place one point of the dividers at 1, in *feet*, and the other among *inches*, when the length is at once seen to be $1' \, 6\frac{1}{2}''$. We mention this mode of numbering, as in some instances we have seen scales with the inches numbered in the opposite direction.

Fig. 21 represents a scale of $\frac{1}{12}$, or 1 inch = 1 foot, showing feet, inches, and half-inches.

Fig. 22 represents a scale of $\frac{1}{24}$, or $\frac{1}{2}$ inch = 1 foot, showing feet and inches.

24. Fig. 23 represents a diagonal scale of $\frac{1}{8}$, or $1\frac{1}{2}$ inches = 1 foot, showing feet, inches, and sixteenths of an inch; the construction of the scale is as follows:—Draw a line about 12 inches long, and divide it into lengths of $1\frac{1}{2}$ inches, which will represent feet; or mark off 6 or 12 inches, and by repeated bisections obtain the required distance. At the

left-hand extremity of this line erect a perpendicular, and along it step sixteen *equal distances of any convenient length;* through each division draw lines parallel to that first drawn. Divide the left-hand foot into twelve equal parts to represent inches; number the scale as shown; and from 0, 1, 2, etc., on the right of $0b$ erect perpendiculars. Draw a line from No. 11 on the bottom line to No. 16 on the perpendicular line, and from each inch in the bottom line draw lines parallel to 11—16; the scale is now complete.

The distance 0—1, on the left of 0, equals 1 inch; $01ab$ is a rectangle, and $0a$ one of its diagonals, divides the fifteen lines between 0—1 and ab. The length of each of the horizontal lines between $0b$ and the diagonal $0a$ differs by $\frac{1}{16}$ of 0—1 from the one next to it above or below; the lengths of these lines are marked on the perpendicular line at 12, counting from the bottom, so that the length of the horizontal line 1 between $0a$ and $0b = \frac{1}{16}$ of 0—1, the one next to it $= \frac{2}{16}$ of 0—1, and so on, each succeeding one being $\frac{1}{16}$ greater. This small division is the $\frac{1}{16}$ of an inch, and the $\frac{1}{192}$ of a foot ($\frac{1}{12} \times \frac{1}{16} = \frac{1}{192}$), as represented by the scale, which is of $\frac{1}{8}$, that is, the distance 0—12 represents a foot, but is, of course, only $1\frac{1}{2}$ inches long. The distance between the two dots on the horizontal line marked 8 is $\frac{8}{16}''$ or $\frac{1}{2}''$; between those on line 5, $1'\ 0\frac{5}{16}''$; and on line 13, $2'\ 7\frac{13}{16}''$.

Fig. 24 represents a diagonal scale of $\frac{1}{12}$, showing feet, inches, and tenths of an inch. The distance between the dots on line 9 is $\frac{9}{10}''$ or $\cdot 9''$; on line 7, $3\frac{7}{10}''$ or $3\cdot 7''$; and on line 3, $1'\ 8\frac{3}{10}''$ or $1'\ 8\cdot 3''$.

Fig. 25 represents a diagonal scale of $\frac{1}{4}$, showing feet, inches, and thirty-seconds of an inch.

25. If now the student compare the two forms of scales illustrated, he will see that it would be impossible to obtain with the plain scale the small divisions we have obtained with the diagonal scale. Though we do not often require such small divisions of a foot, still we often require dimensions differing by, or containing such divisions, and hence the importance of having a scale by which we can readily obtain them.

SECTION VI.

FIRST LINES OF A DRAWING.

26. First Lines of a Drawing.—In making a drawing there are certain lines which are to be drawn before the lines of the object proper are drawn, whether the drawing we are making is from a copy, a sketch of an existing machine, or from the rough sketch of a proposed machine. In all cases there is some line either existing in the object, or which may be considered for convenience as existing in it, about which the object is symmetrical; that line is to be drawn first; generally there are several such lines in the drawing of a machine, and often one principal line. These first lines are called *centre lines*, they should be made very fine and distinct,* and should be drawn with the greatest care, as the correctness of the drawing will in a great measure depend upon their accuracy.

I. *Plane Surfaces.*—In fig. 26 we have an object which is symmetrical with respect to the line *ab;* also each end is symmetrical about *ab* and two other lines; these two lines are *cd* and *ef*. Therefore, before drawing the lines of the object, the lines *ab*, *cd*, and *ef*, should be drawn, and the intersections of these lines taken for the centres of the circles.

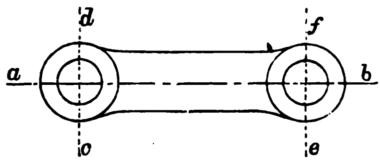


Fig. 26.

We give another example in fig. 27. The object is symmetrical with respect to *ab*, and also to *cd* and *ef;* there are also parts of the object which are symmetrical about *gh*, *kl*, and *mn*. Draw first the triangle ABC, and then bisect its angles, as shown, and the whole of the required centre lines are drawn; the lines of the object may then be put in.

* In the drawings in this book we have used a dotted line, thus —— · —— · —— instead of a fine continuous line, for the sake of distinction and convenience.

II. *Solids.*—In the articles on projection, to which we refer the student, we have shown how solids are represented on a plane by at least two projections, each of which have centre lines common to both. The statements made respecting the drawing of centre lines of plane surfaces apply also to these projections.

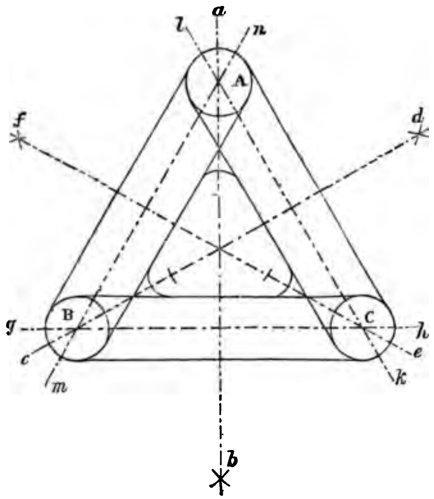


Fig. 27.

Figs. 28 and 29 are two projections of a cylinder, both figures are symmetrical with respect to the line ab, which is therefore to be drawn first. In fig. 28 the line cd, at right

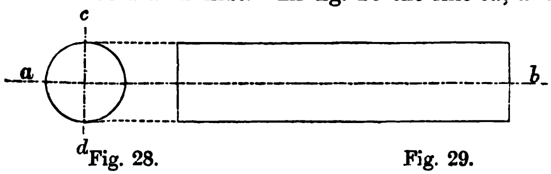


Fig. 28. Fig. 29.

angles to ab, fixes the centre of the circle, but this line is not seen in fig. 29. The line about which the object is symmetrical is its axis, which is represented in fig. 28 by the point of intersection of ab and cd, and in fig. 29 by ab.

CHAPTER II.

SECTION I.

CONSTRUCTIONS FOR CONNECTING CIRCLES AND ARCS OF CIRCLES WITH STRAIGHT LINES, ETC.

27. *To connect two given straight lines containing a right angle by means of an arc of a circle of a given radius.*

Fig. 30.—Let AB and BC be the two given straight lines, and let D represent the given radius.

From B as a centre, with a radius D, cut AB, BC in E and F; through these points draw lines parallel to BC and AB respectively, intersecting in O. Then O is the centre of the required arc of circle (as the angle ABC is a right angle, the arc EF is a quadrant of a circle).

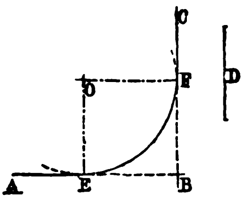


Fig. 30.

28. *To connect two given straight lines which meet, but do not contain a right angle, by means of an arc of a circle of a given radius.*

Fig. 31.—Let AB and BC be the two given straight lines, and let D represent the given radius.

In AB, BC, take any convenient points H and K, and erect perpendiculars HL and KM, equal in length to D. Through L and M draw lines parallel to AB and BC respectively, intersecting in O. Then O is the centre of the required arc of circle. From O draw OE, OF perpendicular to AB and BC respectively, then EF is the length of the arc to be drawn.

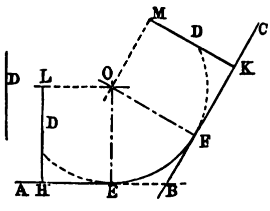


Fig. 31.

29. *To draw a circle to touch three given straight lines.*

Fig. 32.—Let AB, BC, and CD, be the given straight lines. Bisect the angle ABC by the line BH, the centre of the required circle will be in this line. Now bisect the angle BCD by the line CK, the centre of the required circle will be in this line also. Let BH and CK intersect in O. Then O is the centre of the required circle. From O draw perpendiculars meeting AB, BC, and CD in E, F, and G respectively, these are the points of contact between the given lines and the circle.

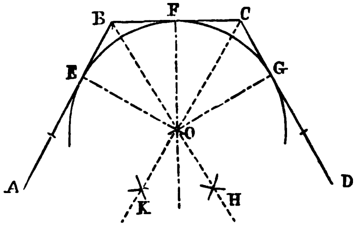


Fig. 32.

30. *To connect a given circle and a given straight line by means of an arc of a circle of a given radius.*

a. When the given straight line passes through the centre of the given circle.

I. *To touch the given circle internally.*

Fig. 33.—Let AB be an arc of the given circle whose centre is O, CD the given straight line, and R the given radius. Draw any radius OP cutting the arc AB in E, and from E set off, inside the circle, EF equal to the radius of the connecting arc R. From O as a centre, with a radius OF, describe the arc HFK cutting the line CD. From any point L in CD draw LM perpendicular to CD, and equal in length to R. Through M draw MK parallel to CD, cutting the arc HFK in K. Then K is the centre of the con-

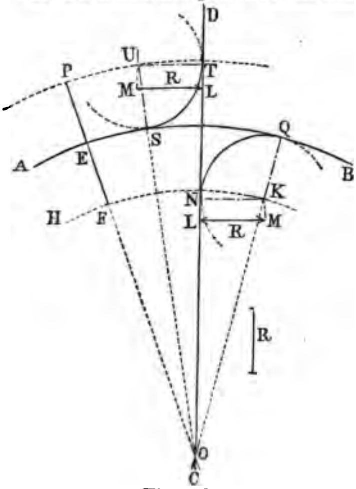


Fig. 33.

necting arc. Draw the radius KN perpendicular to CD, and from O draw the line OK, and produce it to meet AB in Q. Then NQ is the length of the connecting arc.

II. *To touch the given circle externally.*

The construction is shown on the left of the line CD, fig. 33, and is similar to the former case; the only difference is, that the centre U of the connecting arc is outside the given circle. ST is the length of the required arc.

b. When the given straight line does not pass through the centre of the given circle.

I. *To touch the given circle externally.*

Fig. 34.—Let AB be an arc of the given circle whose centre is O, CD the given straight line, and R the given radius. Draw any radius OE and produce it, making EP equal to the given radius R. From O as a centre, with a radius OP, describe the arc PFH cutting the line CD. From any point K, in CD, draw KL perpendicular to CD, and equal in length to R. Through L draw LF parallel to CD cutting the arc PFH in F. Then F is the centre of the connecting arc. Join FO cutting AB in N, and from F draw the radius FM perpendicular to CD. Then MN is the length of the required connecting arc.

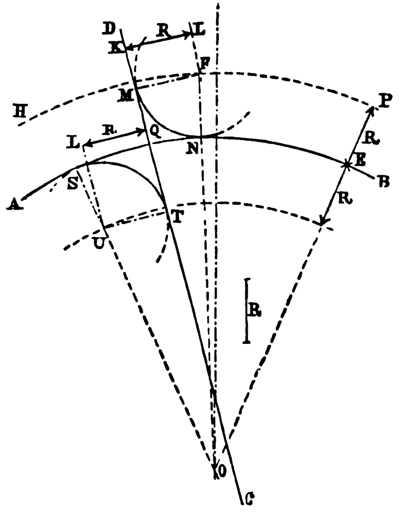


Fig. 34.

II. *To touch the given circle internally.*

The construction is shown on the left of line CD, fig. 34.

31. *To describe a circle to pass through three given points.*

Fig. 35.—Let A, B, and C be the three given points.

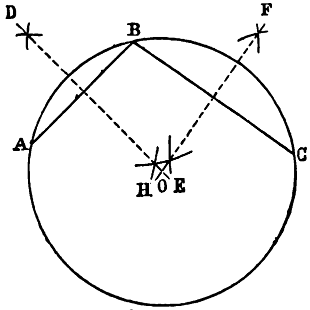


Fig. 35.

Join AB and BC; bisect AB and BC by the perpendicular lines DE and FH, which intersect in O. Then O is the centre of the required circle, and OA its radius.

32. *To draw a tangent to a given circle from a given point.*

a. When the given point is on the given circle.

Fig. 36.—Let ABC be the given circle whose centre is O, and Q the given point. Join OQ and draw DE perpendicular to OQ. Then DE is the required tangent.

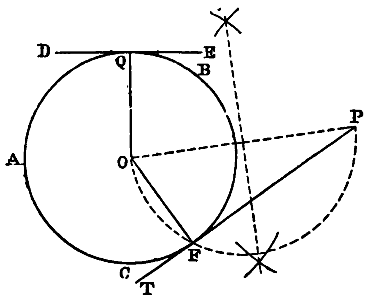


Fig. 36.

b. When the given point is outside the given circle.

Fig. 36.—Let ABC be the given circle whose centre is O, and P the given point. Join PO, and upon it describe a semicircle PFO, cutting the given circle in F. Draw PT; it touches the given circle in F; and is the required tangent.

If FO is joined, PT is perpendicular to FO (the angle in a semicircle is a right angle).

33. *To draw a common tangent to two given circles whose radii are unequal.*

I. *To touch the given circles on the same side of the line which joins their centres.*

Fig. 37.—Let O and P be the centres of the given circles ABC and MNQ. Join OP, and let the line OP cut the given circle ABC in C. From C set off, inside the circle,

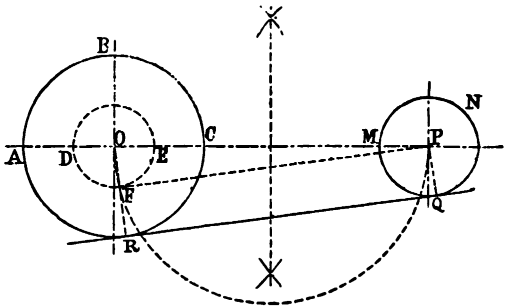


Fig. 37.

CE equal to the radius of the smaller circle MNQ. From O as a centre with a radius OE, describe the circle DEF, and from P draw a tangent PF to this circle by the construction given in the previous figure. Join OF and produce it to meet the circumference of the circle ABC in R. From P draw PQ parallel to OR, meeting the circumference of the circle MNQ in Q. Join QR, which is the required tangent.

II. *To touch the given circles on opposite sides of the line which joins their centres.*

Fig. 38.—The construction is similar to the preceding case, CE = PN is set off outside the circle ABC instead of inside it, as in fig. 37.

Note.—The radius of the circle DEF in fig. 37 is equal to

the *difference*, and in fig. 38 it is equal to the *sum*, of the radii of the two given circles.

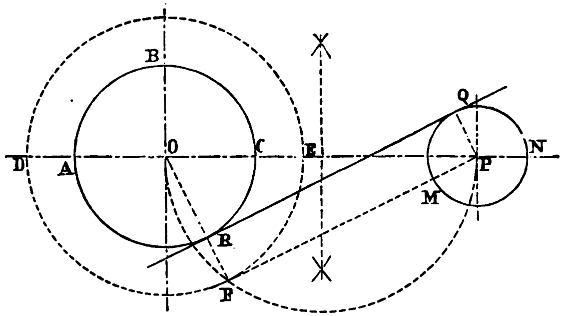


Fig. 38.

34. *To describe a circle which shall touch a given straight line, and also touch a given circle at a given point.*

I. *To touch the given circle externally.*

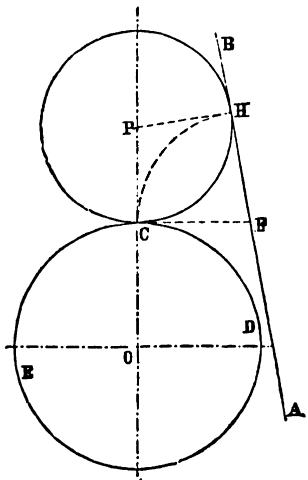


Fig. 39.

Fig. 39.—Let AB be the given straight line, CDE the

given circle whose centre is O, and C the given point. Join OC and produce it, from C draw CF perpendicular to OC, meeting AB in F. From F as a centre with a radius FC, describe an arc of a circle cutting AB in H. Draw HP perpendicular to AB, meeting OC produced in P. Then P is the centre of the required circle, and PH its radius.

II. *To contain the given circle.*

Fig. 40.—The construction is similar to the preceding case, and, as the same letters of reference are used, the student can refer to it.

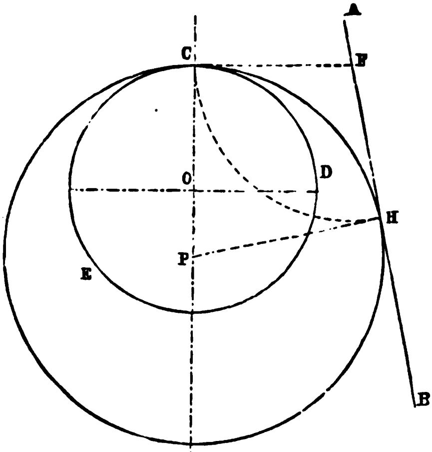


Fig. 40.

35. *To describe a circle which shall touch a given circle, and touch a given straight line in a given point.*

a. When the given straight line does not cut the given circle.

I. *The required circle is to touch the given one externally.*

Fig. 41.—Let CDE be the given circle whose centre is O, and F the given point in the given straight line AB. Through O draw OH perpendicular to AB, meeting the circumference of the circle CDE in H; and from F draw FQ perpendicular

to AB. Join FH, cutting the circumference of the circle CDE in E. Through E draw OE, and produce it to meet the perpendicular FQ in P. Then P is the centre of the required circle, and PE its radius.

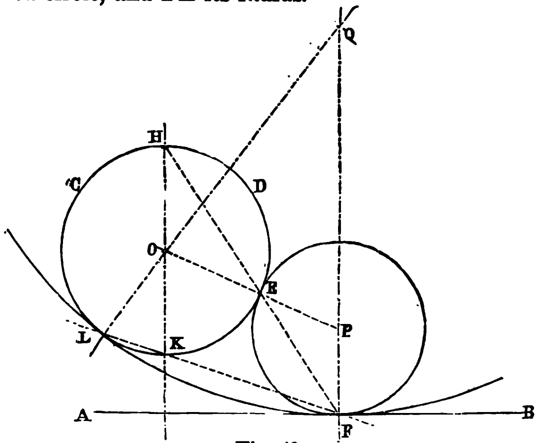


Fig. 41.

II. *The required circle is to contain the given one.*

Fig. 41.—Let CDE be the given circle whose centre is O, and F the given point in the given straight line AB. Through O draw OK perpendicular to AB, cutting the given circle CDE in K; and from F draw FQ perpendicular to AB. Join FK, and produce it to meet the circle CDE in L. From L draw LO, and produce it to meet the perpendicular from F in Q. Then Q is the centre of the required circle, and QL its radius.

b. When the given straight line cuts the given circle.

Fig. 42.—The same construction is employed as in fig. 41, case I.

The following construction may be employed with advantage, and in many cases for drawing purposes it is more accurate.

Fig. 43.—Let CED be an arc of the given circle whose centre is O, and F the given point in the given straight line

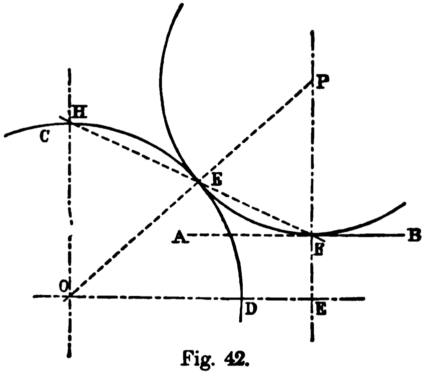


Fig. 42.

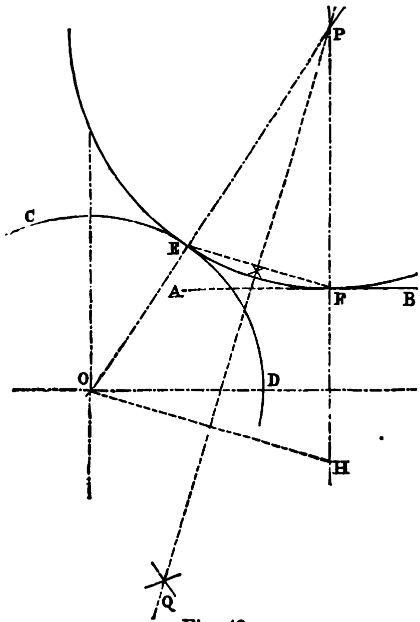


Fig. 43.

AB. Through F draw PFH perpendicular to AB, and from F set off FH, equal to the radius of the given circle CED. Join HO, and bisect it by the perpendicular PQ, cutting HP in P, and from P draw PO, cutting the given circle in E. Then P is the centre of the required circle, and PE its radius.

36. *To describe a circle which shall touch two given circles externally, and touch one of them in a given point.*

Fig. 44.—Let CDE and FHK be the two given circles whose centres are A and B respectively, and let F be the given point. Join FB, and if necessary produce it, making FL equal to the radius of the circle CDE. Join AL, and

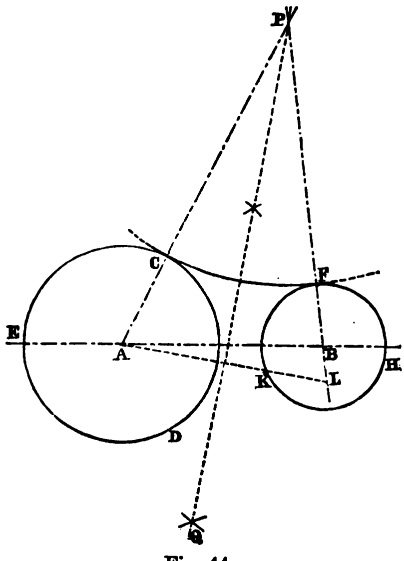


Fig. 44.

bisect it by the perpendicular PQ meeting LF produced in P. Then P is the centre of the required circle, and PF its radius. Join AP cutting the circle CDE in C, then CF is *the length of* the connecting arc of the required circle.

37. *To describe a series of circles to touch each other in succession, and also to touch two given straight lines which are not parallel.*

Fig. 45.—Let AB and CD be the two given straight lines. Produce AB and CD until they intersect in E; if this is inconvenient, draw lines parallel to, and equidistant from, AB and CD intersecting each other. Bisect the angle AEC contained by the lines AB and CD, or by lines parallel to them, by the line EF, which will contain the centres of the required circles; let one of the circles have a given radius R.

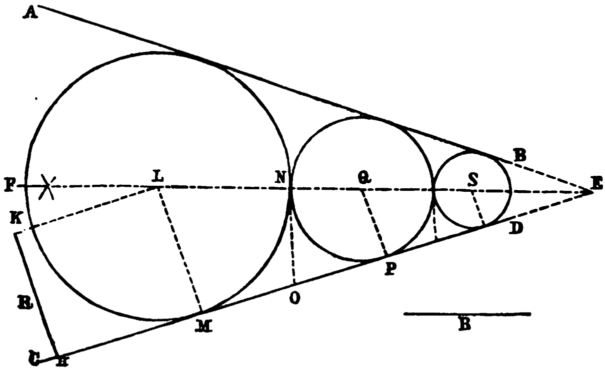


Fig. 45.

At any point H in one of the given lines, as CD, draw HK perpendicular to CD and equal to R; draw KL parallel to CD meeting EF in L. Then L is the centre of the circle FMN of the given radius, and touches the given lines.

Draw the radius LM perpendicular to CD, and from N draw NO at right angles to EF to meet CD in O; from O set off, along CD, OP equal to ON, and draw PQ perpendicular to CD, meeting EF in Q. Then Q is the centre of the second circle which will touch the described circle and the given lines. By repeating this operation, any number of circles can be described on either side of the circle whose centre is L.

38. *To describe a circle which shall pass through a given*

point, and also touch two given straight lines which are not parallel.

Fig. 46.—Let AB and CD be the two given straight lines, and P the given point; let the lines be produced to meet in E. Two circles of different radii can be drawn to fulfil the conditions; one whose centre is between P and E, and the other having its centre on the side of P remote from E.

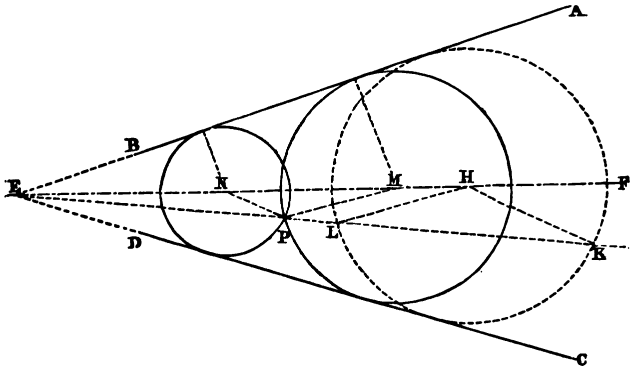


Fig. 46.

Bisect the angle AEC by the line EF, which will contain the centres of the two circles. From any point H in EF describe a circle touching AB and CD, and from E draw a line through P cutting this circle in L and K. If the required circle is to be the larger of the two, join LH, and draw MP parallel to LH cutting EF in M. Then M is the centre of the required circle, and MP its radius.

The centre N of the smaller circle is obtained by drawing PN parallel to KH, as shown by the construction lines.

SECTION II.

DIVISION OF LINES—SQUARE ROOTS OF NUMBERS—AREAS.

39. *To divide a given line into any required number of parts whose lengths shall be in any given proportion.*

Fig. 47.—Let AB be the given line 1·4″ long which is to be divided into, say, three parts in the proportion of ·85, 1·1, and 1·45 (·85 : 1·1 : 1·45).

Draw, from A or B, any line AD, making an angle of about 30° with AB, and upon it set off AC, CE, and EF, equal to ·85, 1·1, and 1·45, respectively, of any convenient unit of length, say an inch. Join FB, and draw E*e* and C*c* parallel to it, then AB is divided in the points *c* and *e* in the required proportion.

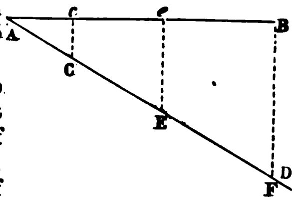


Fig. 47.

A*c* : *ce* : *e*B :: AC : CE : EF.

The length of AF = ·85 + 1·1 + 1·45 = 3·4 = 3·4″, the unit of length being one inch. The line AB is 1·4″ long, and contains 3·4 times some unknown unit of length, of which A*c*, *ce*, and *e*B contain ·85, 1·1, and 1·45, respectively.

40. *To determine by construction the square roots of numbers.*

I. Fig. 48.—Draw a right-angled triangle BAC, whose sides AB, AC are each one unit in length (taken from any scale). Then as AB and AC are each 1, CB will be $\sqrt{2}$. Draw CD at right angles to BC, and equal to AB; join BD. Then BD will be $\sqrt{3}$. In the right-angled triangle BDF, BD is $\sqrt{3}$ and DF (= BC) is $\sqrt{2}$, therefore BF is $\sqrt{5}$. It must be observed that all these roots are expressed in the same unit as AB and AC. The construction just described may be continued to any extent within reasonable limits.

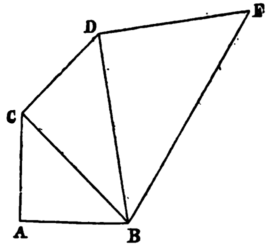


Fig. 48.

The following construction is of more general application than the preceding.

II. Fig. 49.—Let it be required to find the square root of 3·35, and let AB represent the unit of length, say an inch.

Draw an indefinite line EF, and mark upon it AB, AC equal to 1 and 3·35 units, respectively. Bisect BC and describe the semicircle BDC; from A draw AD at right angles to BC meeting the semicircle in D. Then AD will be equal to $\sqrt{3{\cdot}35}$; that is to say, the area of the square described upon AD will be 3·35 times the area of that described upon AB.

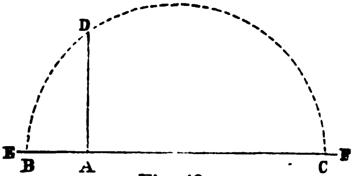


Fig. 49.

The line AD is a mean proportional between AB and AC, which is expressed in the form of a proportion thus,

$$AB : AD :: AD : AC;$$

multiplying the extremes and the means we obtain

$$AD^2 = AB \times AC.$$

The square upon AD is equal in area to the rectangle having AB and AC for its sides.

41. *To determine by construction the square root of a fractional number.*

Fig. 49.—Let it be required to find the square root of $\frac{1}{4}$ AC representing the unit of length. Make AB $= \frac{1}{4}$ AC, and describe the semicircle BDC; from A draw AD at right angles to BC, then AD $= \sqrt{\frac{1}{4}}$. In like manner, if we required the $\sqrt{\frac{3}{4}}$; make AB $= \frac{3}{4}$ AC, then AD $= \sqrt{\frac{3}{4}}$.

42. *To draw a square whose area shall be equal to the sum of the areas of two given squares.*

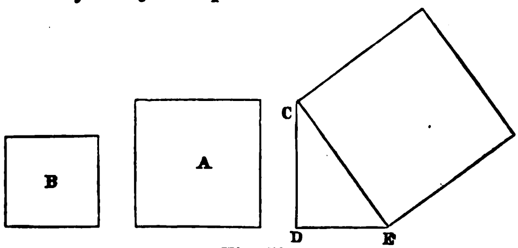


Fig. 50.

Fig. 50.—Let A and B be the two given squares. Con-

struct a right-angled triangle CDE, the sides CD and DE being equal to the sides of the given squares A and B. Then the square described upon CE will be the required square, which may be expressed in the form of an equation, thus:

$$CE^2 = CD^2 + DE^2.$$

$$\text{Let } DE=3, \text{ and } CD=4, \text{ then } CE=5$$
$$\text{for } CE^2 = 4^2 + 3^2$$
$$= 25$$
$$\therefore CE = 5.$$

Note.—This construction can be extended to include circles, and *similar* triangles, parallelograms, and polygons.

Example.—To find the diameter of a pipe equal in area to two given pipes.

43. *To draw a circle whose area shall be equal to the difference of the areas of two given circles.*

Fig. 51.—Let A and B be the two given circles. Draw any line ED equal to the diameter of the circle B, and draw DF at right angles to ED. From E as a centre, with a radius equal to the diameter of the circle A, describe an arc cutting DF in C. Then CD is the diameter of the required circle.

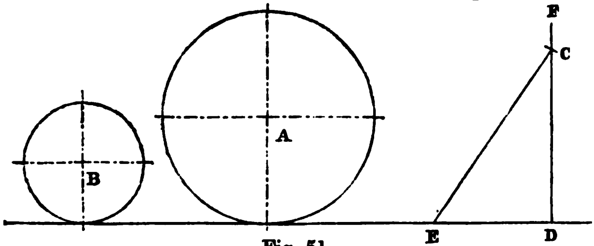


Fig. 51.

Note.—This construction can be extended to include squares, and *similar* triangles, parallelograms, and polygons. In Art. 42, page 40, we described a square equal in area to the *sum* of two given squares, which was expressed as an equation, thus:

$$CE^2 = CD^2 + DE^2.$$

In the present case, taking an example of squares, the obtained square is equal to the *difference* of the two given squares, expressed, thus:

$$CD^2 = CE^2 - DE^2.$$

SECTION III.

THE ELLIPSE.

44. *To determine points in an ellipse, the major and minor axes being given.*

Fig. 52, Plate II.—Let AB and CD be the given major and minor axes (the principal axes, or the diameters of the ellipse, which bisect each other at right angles). From C or D as a centre with a radius AE, equal to one-half the major axis, cut AB in F and H; these points are the *foci* of the ellipse. Between F or H and E take a number of points 1, 2, etc., whose distances from each other decrease as they approach F or H. From F and H as a centre with a radius A1 describe arcs of a circle on each side of AB, and from H and F as a centre with a radius B1 cut these arcs in I, I, these will be points in the ellipse. Then from the same centres with radii A2 and B2 repeat the operation, and thus find the points II, II; and so on with radii A3 and B3, etc., find the points III, III, etc. Through these points draw the half ellipse CAB.

By reversing the describing radii points can be found in the other half, as in the figure, where Fɪᴠ = B4; Hɪᴠ = A4, etc. The number of points to be taken between E and F must depend upon the size of the figure and the degree of accuracy desired.

45. *To determine points in an ellipse by means of a straight edge, when the major and minor axes are given.*

Fig. 53, Plate II.—Let AB and CD be the given axes. Take a slip of paper RS, one of whose edges is a straight line, and mark upon it in fine lines MK equal to AE, and KL to CE; then LM is the difference of the semi-axes. First place the slip of paper so that M coincides with E, and K with B; now gradually change the position of the slip, keeping L and M always upon the axes, until L coincides with E, and K with C. During this change of position of the slip the point K will have traced out one-quarter of the ellipse; the successive positions of K must be marked with a fine pointed pencil, and the curve neatly drawn through the positions. The remaining quarters can be traced in a similar manner; two positions of the slip are shown in the figure.

This is a very correct and simple method of describing an ellipse, and can be employed on drawings without using construction lines as in the preceding figure.

46. *To describe an ellipse by means of a string and a describing point, when the major and minor axes are given.*

Fig. 54, Plate II.—Let AB and CD be the given axes; determine the foci as in Art. 44. At the foci F and H fix pins, to which attach the ends of a thread whose length shall be such that the describing point, as a pencil, shall exactly reach to C or D. In the figure the describing point is shown in two positions P and O. From the construction it will be seen that $DF + DH = OF + OH = AF + AH$ $(= HF + 2AF)$ $= AB$ the major axis. This is the method employed by gardeners for elliptical flower beds.

47. *To draw a tangent and a normal to an ellipse at any point P.*

Fig. 54, Plate II.—Let AB and CD be the axes, and F, H the foci. Join PF, PH, and produce one of them, as FP to G; bisect the angle GPH by the line KL; then KL will be the required tangent. The normal MN is obtained by bisecting the angle FPH, as in the figure. As the normal and tangent are at right angles to each other, if both are required, it is only necessary to bisect one of the angles, as FPH, and draw the tangent at right angles to the normal.

48. *To determine points in an ellipse, any two conjugate diameters being given.*

Fig. 55, Plate II.—Let AB and CD be the given diameters; draw through the extremities A, B, C, and D, of these diameters lines parallel to them, forming a parallelogram KLMN. Divide AN into any convenient number of equal parts, say three; and divide AE into the same number of equal parts, numbering them as shown. Join C1, C2, and D1, D2, and produce D1, D2, to cut C1, C2, in I, II; then I, II, are points in the ellipse, which is completed by drawing a careful arc through them. The construction lines show how the remaining portion of the semi-ellipse is obtained. The other half may be obtained in a similar manner, or by drawing the diameters IEI′, IIEII′, etc., making EI′ = EI, EII′ = EII, etc.

SECTION IV.

THE INVOLUTE OF A CIRCLE—CYCLOIDAL CURVES.

49. The Involute of a Circle.—*If a perfectly flexible line be wound round a circle so as to coincide with it, and be kept stretched as it is unwound, a point in the line will describe a curve called the involute of the circle; the circle is called the evolute.*

To determine points in the involute of a circle.

Fig. 56, Plate III.—Let BCD be a circle, whose centre is A, the involute of which is required. Divide the circumference of the circle BCD into any convenient number of equal parts, say 12, and number them as shown. Draw radii A0, A1, A2, etc.; from the points 1, 2, 3, etc., draw lines, 1I, 2II, 3III, etc., at right angles to these radii, making them equal to the arcs 01, 02, 03, etc. If the arc 01 is small, the chord 01 may be taken and set off along 1I, 2II, 3III, etc., 1, 2, 3, etc., times respectively; or, otherwise, draw a tangent 6VI, and make it equal to the semi-circumference (the radius $\times$ 3·1416). Divide this tangent into the same number of equal parts as there are in the semi-circumference, and make 1I, 2II, 3III, etc., equal to 6I, 6II, 6III, etc., respectively. Through the points, I, II, III, etc., thus determined, draw the curve 0 . . . VI, which is the involute of one-half of the circle. The remaining half may be drawn without further explanation; in the figure the involute of three-quarters of the circle is shown.

The line TT at right angles to 5V is the *tangent* to the involute at the point V; the line 5V is the *normal.*

Note.—The definition given of the involute of a circle can be extended to include any curve by substituting "curve" in place of "circle" in the definition.

50. The Cycloid.—*If a circle rolls along a straight line, and always remain in the same plane, a point in its circumference will describe a curve called the cycloid. The straight line is the director, the circle the generating circle, and the point the generator.*

To determine points in a cycloid, the director and the generating circle being given.

Fig. 57, Plate III.—AB is the director, and D06 the generating circle whose centre is C. A line DE parallel to AB and drawn through C will be the path of the centre of the generating circle as it rolls along AB. Divide the circumference of the circle D06 into any convenient number of equal parts, say 12, and number them as shown. From any point 0 in AB mark off 0—6 equal to the semi-circumference of the circle by either of the following methods:—

I. By stepping off the *chord* 01 of the circle six times.

II. By calculation make 0—6 on AB equal to the radius of the circle × 3·1416. Divide 0—6, on AB, into six equal parts; from each of the points, 1, 2, 3, etc., erect perpendiculars meeting DE in 1, 2, 3, etc.; from these points as centres with C0 as radius describe arcs of circles. From 1, 2, 3, etc., on the circle D06 draw lines parallel to AB cutting the arcs described from corresponding centres in DE, in I, II, III, etc.; through these points draw the curve 0 . . . VI, which is one-half of the required cycloid. As the other half is exactly similar, it can be drawn without further explanation.

From the construction it will readily be seen that the chords 1I, 2II, 3III, etc., are equal to the chords 01, 02, 03, etc., so that the points in the curve may be obtained by setting off these chords from 1, 2, 3, etc., in AB. The points 0, I, II, etc., are points in the curve corresponding to the positions 0, 1, 2, etc., of the centre of the describing circle.

The error introduced into the construction by using method I. is small, providing the chord taken is small; see Art. 10, page 14. The line TT at right angles to the chord 3III is the *tangent* to the cycloid at the point III; the chord 3III is the *normal* at that point.

51. The Trochoid.—Referring to Art. 50—*If the generator is not in the circumference of the generating circle, the curve traced will be a trochoid.*

To determine points in a trochoid, the director, the generating circle, and the generator being given.

Fig. 58, Plate III.—AB is the director, D06 the generating circle whose centre is C, and P the generator, in the first case taken within the circumference of the circle. Proceed as in the last figure and obtain the points 0, 1′, 2′, etc., corresponding to 0, I, II, etc., in that figure; join these points with 0, 1, 2, etc., in DE. Upon the radii 00, 11′, 22′, etc., mark off 00, 1I, 2II, etc., equal to CP. Through the points 0, I, II, etc., thus obtained draw the curve 0 . . . VI, which is one-half of the required trochoid. This curve is the path of the centre of the crank pin of a locomotive crank, where D06 is the driving wheel, and P the centre of the crank pin.

If the describing point is without the circumference of the generating circle, the trochoid is of the form shown by the outer curve QQQ, fig. 58. The construction of the curve is similar to the preceding case, the radii 00, 11′, 22′, etc., being produced and made equal to C*q*, where *q* is the describing point; through the extremities of these radii the curve is to be drawn.

52. The Epicycloid.—*If a circle rolls along the outside of a circle, and always remain upon the same plane as the circle upon which it rolls, a point in its circumference will describe a curve called an epicycloid. The rolling circle is the generating circle, the circle upon which it rolls the director, and the point the generator.*

To determine points in an epicycloid, the director and the generating circle being given.

Fig. 59, Plate III.—ABC is the director, S its centre, and KLM the generating circle whose centre is D.

A circle DEF described from S as a centre with a radius SD will be the path of the centre of the generating circle as it rolls along ABC. Join SD and produce it to meet the circle KLM in K; divide the semicircle OMK into any convenient number of equal parts, say 6, and number them as shown. From any point 0 in ABC make the arcs 01, 12, 23, etc., equal to the arcs 01, 12, 23, etc., of the semicircle OMK, by stepping along ABC a small chord, as 01, of the generating circle. From S draw radii S1, S2, S3, etc., and and produce them to meet DEF in 1, 2, 3, etc.; from these points as centres with D0 as radius describe arcs of circles.

Make the chords 1I, 2II, 3III, etc., equal to the chords 01, 02, 03, etc.; through the points 0, I, II, etc., describe the curve 0 . . . VI, which is one-half of the required epicycloid. The other half being exactly similar can readily be drawn.

As the director is a circle generally not of the same diameter as the generating circle if the chords 01, 02, etc., in ABC are made equal to corresponding chords in KLM, the arc 060 of ABC will not be equal to the circumference of KLM unless the circles are of equal diameter. The following will be a more correct construction.

The circumferences of circles bear the same ratio to each other as their diameters; therefore, if the diameter of the generating circle is one-half that of the director, the epicycloid will occupy one-half of the director; if the ratio of the diameters is $\frac{1}{3}$, the curve will occupy $\frac{1}{3}$ of the director, and so on. Therefore, when the ratio of the diameters is a simple one, as just represented, the arc 060 of ABC can easily be determined and then divided into the same number of equal parts as the generating circle. If the ratio is not a simple one, make the angle 0S0 such that it bears the same ratio to four right angles as the diameter of the circle KLM bears to the diameter of the circle ABC. For example, let KLM be 1″ in diameter, and ABC 3·6″; the ratio is $\frac{1}{3\cdot6}$, therefore the angle $0S0 = \frac{360°}{3\cdot6} = 100°$. Having drawn the angle 0S0, proceed to describe the curve as previously stated.

The line TT at right angles to the chord 3III is the *tangent* to the epicycloid at the point III; the chord 3III is the *normal* to the same point.

53. The Hypocycloid.—*If a circle rolls along the inside of a circle, and always remain in the same plane as the circle upon which it rolls, a point in its circumference will describe a curve called a hypocycloid. The rolling circle is the generating circle, the circle upon which it rolls the director, and the point the generator.*

To determine points in a hypocycloid, the director and the generating circle being given.

Fig. 59, Plate III.—ABC is the director, S its centre, and NOP the generating circle whose centre is H. The con-

struction is similar to that employed for the epicycloid; the centres of the rolling circle are marked H, 1′ 2′, etc.; and the obtained curve is numbered 0 . . . VI . . . 0. The same remarks made respecting the setting-out of the path 0S0 on ABC for the epicycloid applies also for the hypocycloid.

If the diameter of the generating circle is equal to the radius of the director, the hypocycloid becomes a diameter of the director; this property will be considered in the articles on the teeth of wheels.

The cycloidal curves described in Arts. 50, 52, and 53, and the involute of a circle described in Art. 49, are employed for the teeth of wheels; see the paragraphs on teeth of wheels in a succeeding chapter.

CHAPTER III.

SECTION I.

PROJECTION—PROJECTIONS OF POINTS AND LINES.

54. Projection.—The terms "a drawing" and "a plan," as applied to the picture or representation of an object, as of a machine or a portion of one, are pretty well understood in a general sense; but, however, not so clearly as to warrant the omission of a definition of such terms in this book; more especially as there exists some difference of opinion respecting this part of the subject. In the articles that follow we shall lay down those principles of drawing that we consider of the greatest importance for a clear and systematic treatment of the subject; and upon these principles we shall execute all drawings. The principles stated will apply to all the figures in the book, but the examples that immediately follow will be of a simple character; the more difficult cases will be treated in subsequent articles.

55. A drawing of an object, as a machine, consists of a representation of that object on a plane surface, as a level sheet of paper; this representation is made up, if the object is a solid, of at least two separate figures or views as seen by the observer from two distinct positions; each of these views is termed a *projection*. We shall now proceed to define this term as applied to a point, a line, a surface or plane, and a solid.

For the purpose of fully representing the form of solid objects, the projections are made upon two distinct planes which are assumed to be at right angles to each other, and are termed the horizontal and vertical planes respectively;

the former being represented by say a level table, or a drawing-board in its usual horizontal position, the latter by a wall parallel to the table. These planes are called *co-ordinate planes*, and the projections *orthographic projections.* Fig. 60 represents (in isometrical projection) the two planes; ABCD is the horizontal, and ABEF the vertical plane.

56. Projections of a Point.—*The projection of a point upon a given plane is the foot of the perpendicular let fall from that point upon the plane.*

Let H, fig. 60, represent a point raised above the horizontal, and in front of the vertical plane. From H let fall a perpendicular H*h*, meeting the horizontal plane in the point *h*, then *h* is the projection of H upon the horizontal plane ABCD. From H draw a perpendicular, H*h′*, to the vertical plane, meeting it in *h′*, then *h′* is the projection of H upon the vertical plane ABEF.

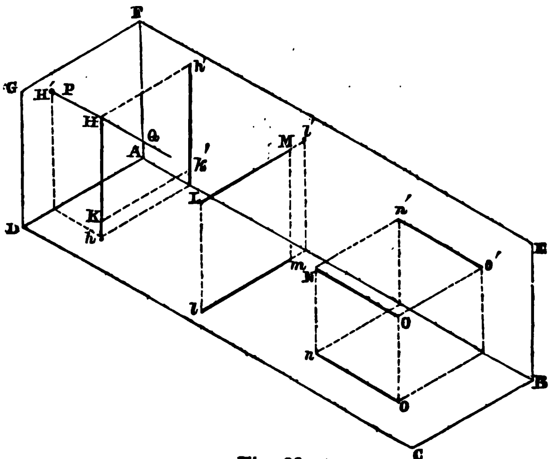


Fig. 60.

A projection upon a horizontal plane is termed a *plan*, and one upon a vertical plane an *elevation;* therefore, *h* is the plan, and *h′* the elevation of the point H with respect to the given planes ABCD, ABEF.

57. Projections of a Line.—*The projection of a line upon a given plane is the line containing the projections of all the points of that line.*

I. Let HK, fig. 60, represent a straight line in front of the vertical, and perpendicular to the horizontal plane above which it is raised. Let the line HK be produced meeting the horizontal plane in h, then h is the projection of the line HK upon the plane ABCD. From H and K draw perpendiculars, Hh' and Kk', to the vertical plane meeting it in h', k'; join $h'k'$, then $h'k'$ is the projection of the line HK upon the plane ABEF. The projection h is the plan, and $h'k'$ the elevation of the line HK; that is, they are the projections of the line HK upon the horizontal and vertical planes respectively.

II. Let LM, fig. 60, represent a straight line above the horizontal, and perpendicular to the vertical plane, but not touching the latter. Let the line LM be produced meeting the vertical plane in l'; and from L and M let fall perpendiculars, Ll and Mm, meeting the horizontal plane in l and m; join lm, then l' is the elevation, and lm the plan of the line LM.

III. Let NO, fig. 60, represent a line parallel to both planes and touching neither of them, then no is the plan, and $n'o'$ the elevation of the line.

As the projections of a straight line are straight lines, it is sufficient to obtain the projections of two points in the line, as the extremities, for each projection, through which the projection is to be drawn. It often happens that the direction of the projection is known, then one point in the projection is sufficient.

58. Before proceeding to the other cases we will show how these principles are applied in practice. As before stated, our drawings are made upon a simple sheet of paper, the surface of which is a plane. Let DCEF, fig. 61, represent the two planes ABCD, ABEF in fig. 60, one of which is turned through a right angle so that they form one plane, as a sheet of paper. Then the plans and elevations of the lines HK, LM, and NO, shown in fig. 61, are in the positions they would occupy in our drawing. The line AB which divides the two planes, and, generally, separates the plan from the

elevation, is termed the *ground line.* The ground line, besides representing the *line of intersection* of the two planes, also represents the elevation of the horizontal plane when we refer to the elevation of an object; and when we refer to the plan of an object, it represents the plan of the vertical plane. This may readily be seen by referring to fig. 60, where AB is the projection of FE upon the horizontal plane, and of DC upon the vertical plane. Therefore the distance of a point or of a line from AB, measured towards DC, represents the distance of that point or line from the vertical plane; and, similarly, if the distance is taken from AB towards EF it represents the distance of the point or line from the horizontal plane.

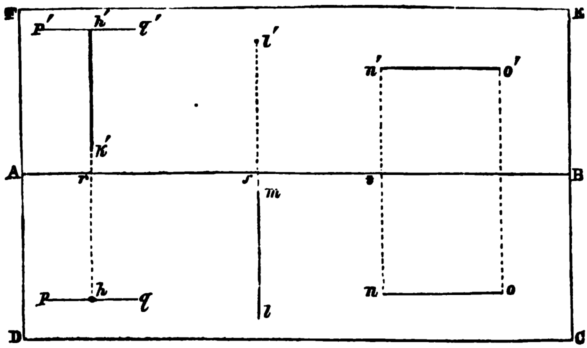


Fig. 61.

59. To draw the Projections of a Point.—Suppose we wish to draw the plan and elevation of a point H which is ·7″ in front of the vertical, and ·75″ above the horizontal plane.

Let DCEF, fig. 61, be our sheet of paper, which we shall always assume as rectangular. Take any point A, about half way between D and F, and through it draw the ground line AB parallel to DC. Draw lines pq, $p'q'$ parallel to and at distances of ·7″ and ·75″ respectively from AB; then the plan of H will be in pq, and the elevation in $p'q'$. Let h represent the plan; from h draw hh' perpendicular to AB meeting $p'q'$ in h' then h' is the elevation of H.

The projections, h and h', of the point H fix the position of H with respect to the co-ordinate planes; if the projections

are not given, but simply the distance of the point from each of the two planes, then the point is not definitely fixed, as it may be anywhere in the line PQ, fig. 60, passing through the point and parallel to the co-ordinate planes. In the example just given we took *h* in *pq* as the plan of H and obtained the elevation from it. Let ADGF, fig. 60, be another plane at right angles to the co-ordinate planes, and H′ the projection of H upon it. Then the position of H is wholly determined if we state its position relatively to the three planes.

60. In the class of drawing we shall require in machine construction the object is assumed as resting on a horizontal plane, and its position with respect to the vertical plane is determined by other considerations than those named in the previous article. The third plane ADGF is generally assumed as passing through the centre of the object, and becomes the *centre line* of the drawing, to which we shall often have occasion to refer.

As every object which we have to represent, or "make a drawing of," is made up of surfaces, the boundaries of which are lines, and as, in most cases, these surfaces are the boundaries of solids, it is obvious that if we are able to draw the projections of lines under different conditions of position with respect to the co-ordinate planes, the projections of simple solids will present little difficulty. And as the student advances he will be able to draw with comparative ease the most difficult objects we meet with in machine construction.

61. To draw the Projections of a Straight Line.—A line ·7″ long is to be represented in plan and elevation according to the following six positions, in which we shall assume its position with respect to the third plane to be fixed arbitrarily:—For cases I., II., and III., see fig. 61, and also fig. 60; for IV., V., and VI. see fig. 62. If a line is called AB, its plan is marked *a b* and its elevation *a′ b′*.

Fig. 61.—I. The line, HK, is in front of the vertical plane, and ·7″ from it; and perpendicular to the horizontal plane above which it is raised ·1″. Take any point *r* in the ground line AB, and draw through it a perpendicular *hh′*; make *rh* equal to ·7″, and *rk′*, *rh* equal respectively ·1″, ·8″; join *h′k′*. Then the point *h* is the plan; and the line *h′k′* the elevation of the line HK.

II. The line, LM, is parallel to the horizontal plane and ·7″ above it; in front cf the vertical plane and ·1″ from it. Take any point *s* in AB and draw through it a perpendicular *ll′*; make *sl′* equal to ·7″, and *sm*, *sl* equal respectively ·1″, ·8″; join *lm*. Then *l′* is the elevation, and *lm* the plan of the line LM.

III. The line, NO, is parallel to both planes; ·5″ above the horizontal, and ·6″ in front of the vertical plane. Take any point *t* in AB and draw through it a perpendicular *nn′*; make *tn′*, *tn* equal respectively to ·5″ and ·6″. From *n* and *n′* draw *no*, *n′o′* each equal ·7″; then *no* is the plan, and *n′o′* the elevation of the line NO.

Fig. 62.—IV. A line, RS, parallel to the vertical plane but not touching it, and inclined to the horizontal at an angle of 30°, is shown in plan and elevation by *rs*, *r′s′*, respectively. The drawing of this and the following case is obvious from the figure, we shall therefore pass on to the remaining case.

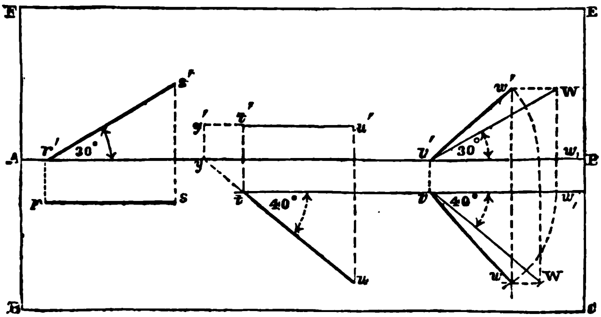


Fig. 62.

V. A line, TU, parallel to the horizontal plane but not touching it, and inclined to the vertical at 40° is shown in plan and elevation by *tu*, *t′u′*, respectively.

VI. A line, VW, inclined to both planes is shown in plan and elevation by *vw*, *v′w′*, respectively. The line VW is inclined to the horizontal plane at 30°, and to the vertical at 40°.

1*st Construction in Vertical Plane.*—Take any point *v′* in AB and draw *v′*W inclined to AB at 30°; from W draw

Ww_1 perpendicular to AB and meeting it in w_1. Then $v'w_1$ would represent a plan of v'W. Let the plan of the end V of the line be below AB, that is, in front of and not touching the vertical plane. From v' draw $v'v$ perpendicular to AB, and fix upon the point v; through v draw vw_1 parallel to AB.

2nd Construction in Horizontal Plane.—Draw on the horizontal plane vW equal to v'W at an angle of 40° with v_1w_1; from v as a centre with a radius vw_1 describe an arc of a circle meeting a line drawn from W and parallel to AB in w. Now join vw; from w draw ww' at right angles to AB meeting a line drawn from W (above AB) parallel to AB in w'; join $v'w'$. Then vw is the plan, and $v'w'$ the elevation of the line VW. We may either obtain the plan and then project the elevation from it, as explained, or *vice versa,* as shown in the figure. This problem is required in projecting *shadows.*

SECTION II.

PROJECTIONS OF PLANE AND SOLID OBJECTS.

62. Projections of a Plane Figure.—*The projection of a plane figure upon a plane is the figure made up of the projections of all the lines of that figure.*

Let HKON, fig. 63, Plate IV., represent a rectangular surface in front of the vertical, and above the horizontal plane; the lines HN, KO, and therefore the surface being perpendicular to the latter plane. Produce HN and KO meeting the horizontal plane ABCD in h, k; join h, k. Then hk is the projection of the surface HKON upon the plane ABCD. From H, K, O, and N draw perpendiculars to the vertical plane ABEF meeting it in h', k', o', and n'; join these points. Then $h'k'o'n'$ is the projection of HKON upon the plane ABEF. The projection hk is the plan, and $h'k'o'n'$ the elevation of HKON.

63. Projections of a Solid.—*The projection of a solid upon a plane is the figure made up of the projections of all the surfaces of that solid.* The projections of all the surfaces are not seen in any one projection of the solid, but it is often

necessary to represent them in the drawing; in which case such surfaces are represented by dotted lines. We shall refer to this later on.

Let HKL..., fig. 63, Plate IV., represent a prism whose base is a square, we will call it for convenience Y. In the position indicated the solid is in front of the vertical plane and above the horizontal; its faces are parallel and perpendicular to the co-ordinate planes. Required its projections.

Produce the lines HN, KO, etc., meeting the horizontal plane ABCD in h, k, l, and m; join these points. Then $hklm$ is the projection of Y upon the plane ABCD. From H, K, O, and N draw perpendiculars to the vertical plane ABEF, meeting it in h', k', o', and n'; join these points. Then $h'k'o'n'$ is the projection of Y upon the plane ABEF. The projection y is the plan, and y' the elevation of Y.

64. To Draw the projections of a Plane Figure.—A rectangle HKON, fig. 63, Plate IV., whose position with respect to the co-ordinate planes is the same as that stated in Art. 62, page 55, is to be represented in plan and elevation. The rectangle is $1·5'' \times 1''$; it is $1·5''$ above the horizontal plane, and $3·25''$ in front of the vertical. The figures are drawn to a scale of $\frac{1}{4}$.

Fig. 64, Plate IV.—Take any point a in the ground line AB, and draw through it a perpendicular hh'; make an', ah' equal respectively, $1·5''$, $3''$, and ah equal $3·25''$. Upon $n'h'$ describe the rectangle $h'k'o'n'$, which is the elevation. From h draw hk parallel to AB meeting a perpendicular $k'k$ in k, then hk is the plan of the rectangle HKON.

If $hklm$, $h'k'$ are plan and elevation of a square, we should draw the plan first, and project the elevation from it. As a rule, that projection of an object which shows its *true* form, that is, a projection upon a plane parallel to the plane of the object, should be drawn first, and the others projected from it.

65. To draw the Projections of a Solid.—A prism HKL ..., fig. 63, Plate IV., whose position with respect to the co-ordinate planes is the same as that stated in Art. 63, page 55, is to be represented in plan and elevation. The prism is $1·5''$ high, its base is a square of $1''$ side; and its distances from the horizontal and vertical planes are $1·5''$, $2·25''$, respectively. The figures are drawn to a scale of $\frac{1}{4}$. Draw

a line *ml*, fig. 64, Plate IV., 2·25″ from AB, and upon it describe the square *hklm*, which is the plan of the prism. From *h*, *k* draw *hh′*, *kk′* perpendicular to AB; make *an′*, *n′h′* each equal 1·5″. Upon *n′h′* describe the rectangle *h′k′o′n′*, which is the elevation of the prism.

66. In Art. 58, page 51, we stated that one of the co-ordinate planes is turned through a right angle, so that the two planes form one plane; it is usual to assume the vertical plane to be turned down to coincide with the horizontal plane, but it is quite immaterial which we turn down. To illustrate this we have shown in figs. 63 and 64, Plate IV., a prism Y; its plan is marked *y*, and its elevation *y′*; and in figs. 65 and 66 a similar prism marked Z, *z*, and *z′*. Figs. 63 and 65 show the planes and the prisms in isometrical projection; figs. 64 and 66 show them in orthographic projection.

If we assume the vertical plane to be turned down, then the prism would occupy the position shown in fig. 67, which is an end elevation of the planes and the object; BC is the horizontal plane, and BE the vertical. The horizontal projection is made upon BC, fig. 67, and the vertical one upon BE, which is then turned through a right angle into the position BE_1. The projections are shown in fig. 64 in the positions they would occupy in our drawing; ABCD is the horizontal plane, and ABE_1F_1 the vertical; *y* the plan and *y′* the elevation of the prism Y.

If now we assume the horizontal plane to be turned through a right angle, keeping the vertical plane fixed; the projections would occupy the same relative positions as before, as shown in figs. 66 and 68. AF, fig. 68, is the vertical plane, and AD the horizontal, which is turned into the position AD_1. The projections are shown in their usual positions in fig. 66; ABC_1D_1 is the horizontal plane and ABEF the vertical; *z* the plan and *z′* the elevation of Z.

The plans and elevations in figs. 64 and 66 are similar in every respect; it is therefore immaterial which of the co-ordinate planes we assume as turned to coincide with the other, but it is usual to consider the horizontal plane as the fixed one. The relative positions of the two planes before and after rotation is shown more clearly in figs. 63 and 65, and as the same letters of reference are used for these as for

figs. 64, 66...68, the student can refer to the description of the latter figures. The plan and elevation of Y, after rotation of the *vertical* plane, are shown in fig. 63 by $hklm$, $h'_1k'_1o'_1n'_1$, respectively. The plan and elevation of Z, after rotation of the *horizontal* plane, are shown in fig. 65 by $p_1q_1r_1s_1$, $p'q't'u'$, respectively. To understand this part of the subject thoroughly we advise the student to make a cardboard model of the planes which will greatly assist him in his study.

SECTION III.

THE TERMS PLAN, ELEVATION, AND SECTION—POSITIONS OF THE VARIOUS PROJECTIONS.

67. Plan, Elevation.—There are several terms used to denote the different projections of an object which we will now explain; and we may observe that there is some difference of opinion respecting the relative positions of the projections of an object. In the present book we have adopted that which we consider to be the simplest and most natural disposition of these projections, and we request the student to pay particular attention to the explanation given, as he will then have no difficulty in determining the position for any projection of an object he may have to make.

68. Figs. 69...72.—These figures represent in elevations and plan a prism, whose base is a square, resting on the horizontal plane; on three faces are placed small prisms differing in length and height, producing an object not symmetrical, as better suited for our purpose.

Fig. 69 is a *Plan*, that is, a projection upon a horizontal plane ABCD. Fig. 70 is an *Elevation*, that is, a projection upon a vertical plane ABC″D″. Figs. 71 and 72 are also elevations.

Fig. 70 is a *Front* or *Longitudinal Elevation;* figs. 71 and 72 are *End* or *Side Elevations.* The term "view" is sometimes used instead of "elevation."

These terms are applied to the representations of the object as a whole, and not with respect to its individual parts; for

Fig. 71. Fig. 70. Fig. 72.

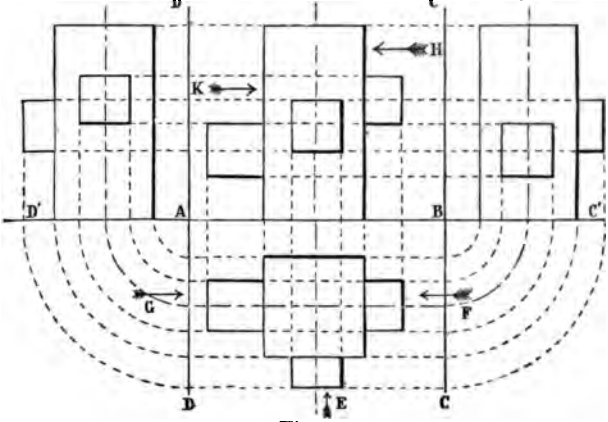


Fig. 69.

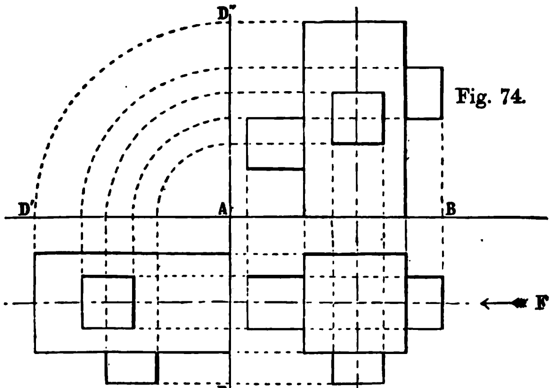


Fig. 74.

Fig. 75. Fig. 73.

instance, we may have in one projection of a machine *nuts* in various positions, as plans and elevations, but these latter terms apply only to the individual parts and have nothing whatever to do with the object as a whole. Again, the terms plan and elevation of an object apply simply to the one fixed position of that object; that is, if we change the position of the object then the plan and elevation change also, and the elevation may become the plan, and the plan the elevation.

69. Section.—*If an object is cut by a plane the surface made by the cutting plane is called a section; for every section thus made we have a second similar section.* Sections are made for the purpose of representing parts of an object, or a combination of objects, which could not otherwise be seen. In making our drawings we generally project the section upon a plane parallel to the section plane so as to show the true form of the section; if the section contains many separate pieces or objects, then we make a section of the whole according to circumstances, and of course some of the pieces cut by this common section plane may not have their true form represented in the projection. Generally, we have data which enables us to represent the sections of objects without actually making the sections; we therefore assume the section as made, and proceed to make our drawings from the known data.

The disposition of the various sectional projections is the same as if they were ordinary projections; following the order of Art. 68 we have *Sectional Plan*, *Longitudinal Sectional Elevation*, and *Sectional End Elevation* or *Cross Section*. We shall frequently use these terms, and to give the student a few simple examples of them refer him to figs. 81, 82, 83, and 84. Sectional surfaces are usually distinguished from other surfaces by drawing diagonal lines across them, as in figs. 154, 155, etc.; other examples may be seen in the working drawings accompanying this work.

70. Positions of the Projections.—In the articles on projection we have defined what a projection is; in Art. 63, page 55, we have defined the projection of a solid, and in Art. 65 shown how to draw the plan and elevation of that solid. We now wish to show that our disposition of the several projections of an object is consistent with the definition of projection we have given. Suppose an object, the

prism before given, rests upon a horizontal plane ABCD; if now we let fall perpendiculars from every point of the object upon that horizontal plane (or any plane parallel to it), we obtain a projection of the object, which is called a plan. The eye of the observer when looking at the object in the position named is *above* the object, and the plane *below* it, so that we have these three in the following order: observer, object, projection, the object coming between the observer and the plane of projection. If now we wish to have a vertical projection of the object as seen from the position E, fig. 69, we erect a vertical plane at any convenient distance on the opposite side of the object to that occupied by the observer, say at AB, and project an elevation upon it, then turn the vertical plane through a right angle so that it coincides with the horizontal plane; the vertical plane will then be represented by ABC″D″, and the elevation of the object by fig. 70. Suppose we wish to see the form of the object, as seen from a position F; erect a vertical plane, as AD, and project an elevation upon it, then, for convenience, turn this new vertical plane into the position AD′, so as to coincide with the original position of the vertical plane AB. Then turn it down through a right angle so that it shall coincide with the altered position of the vertical plane AB, and with the horizontal plane ABCD; the position of this new elevation will obviously be that shown by fig. 71.

Again, suppose we wish to have an elevation of the object as seen from G; erect a vertical plane at BC, and project an elevation upon it; turn the plane into the position BC′, and then turn it to coincide with the horizontal plane and we have this new elevation as shown in fig. 72.

71. We trust these examples will be sufficient to clear up any difficulty the student may meet with in ordinary projections; but before leaving this part of our subject we must refer to figs. 73...75, as they exhibit a disposition of the projections which is sometimes employed. Figs. 73 and 74 occupy the same relative positions as figs. 69 and 70; fig. 75 is exactly similar to fig. 71, but its position relatively to fig. 73 is not the same as that of fig. 71 to fig. 69. Instead of turning the plane AD to coincide with AB and then turning it down along AD′ into the horizontal plane, it is turned from

its first position AD into the horizontal plane, giving the elevation as shown by fig. 75.

The different changes of the planes which we have enumerated do not really take place in our actual drawing, but they must be considered mentally by the draughtsman as if they did take place; a little practice will enable the student to dispose of any difficulty that may arise under this head.

72. In the previous articles we have assumed the existence of an object from which we obtained the different projections, but in designing a machine we do not possess such an object, for it is our aim to produce drawings from which the machine can be constructed. However, so far as regards the disposition of the projections, it is immaterial whether the object exists or not, as the same order is observed in both cases. If we have a plan, as fig. 69, and an elevation, as fig. 70, and we wish for another elevation, such as would be seen by looking at the object from a position F, fig. 69, or H, fig. 70, we project the new elevation as shown by the construction lines, and place it in the position shown by fig. 71. If we require an elevation as would be seen from G, fig. 69, or K, fig. 70, we project it as shown by the construction lines and obtain fig. 72, which is the required elevation. The construction lines fully explain how fig. 75 is obtained from figs. 73 and 74.

If the longitudinal elevation is of considerable length and we require an end elevation as seen from H, fig. 70, or a cross section made by a plane near to the end H, then we may in exceptional cases place the required elevation or section in the place occupied by fig. 72, instead of that occupied by fig. 71; but it must be observed it is an exception made for convenience only, and should not be employed but under exceptionable circumstances.

SECTION IV.

EXTENSION OF THE PLANES.

73. Hitherto we have considered the horizontal and vertical planes as two surfaces at right angles, and terminating in one

direction in a common line, the line of intersection AB, and indefinite in the other direction. With such planes all our plans would appear below AB, and elevations above it. But there are cases in which one projection is represented in a drawing as crossing another; that is to say, a portion of a plan may be above AB, or a portion of an elevation may be below AB, or a portion of each may be on the side AB opposite to that which it usually occupies; and further, the plan and the elevation may change places. We will now proceed to explain how this happens. Instead of our planes terminating in AB, we shall take two planes intersecting in a common line AB and extending indefinitely in both directions, as shown in figs. 76...78, Plate IV.

Figs. 76...78, Plate IV.—In fig. 76, CDD_1C_1 is the horizontal plane, and EFF_1E_1 the vertical plane; this figure is in isometrical projection. Fig. 77 is an end elevation of the planes, showing the relative positions of a number of lines with respect to those planes; DD_1 is the horizontal plane, and FF_1 the vertical plane. Fig. 78 is a front elevation of the planes, with the plans and elevation of the lines as they are represented in our drawings, and in the following examples this figure must be referred to for plans and elevations. In this figure we have assumed, for convenience, the vertical plane to be fixed, and the horizontal plane to be turned to coincide with it.

74. Let HK represent a line *above*, perpendicular to, and touching, the horizontal plane, and in *front* of the vertical plane, then h is its plan, and $h'k'$ its elevation.

Let LM represent a line *below*, perpendicular to, and touching, the horizontal plane, and in *front* of the vertical plane, then l is its plan and $l'm'$ its elevation (l is in $l'm'$).

Let NO represent a line *above*, perpendicular to, and touching, the horizontal plane, and *behind* the vertical plane, then n is its plan, and $n'o'$ its elevation (n and n' coincide).

Let PQ represent a line *below*, perpendicular to, and touching, the horizontal plane, and *behind* the vertical plane, then q is its plan, and $p'q'$ its elevation.

Let RS represent a line inclined to both planes, which passes *below* the horizontal plane, and *behind* the vertical, then rs is its plan, and $r's'$ its elevation.

75. In the above examples we have for every change of position of the object, with respect to the planes, a change in the position of its plan and elevation. It will not be difficult to account for this if the student has a clear idea of projection, of the position of the planes, and of the change of their position, which enables us to represent them on a sheet of paper. It will be seen by referring to fig. 78 that the *plans* of two of the lines, NO and PQ, are above AB; and the *elevations* of LM and PQ are below AB.

SECTION V.

TRACES OF LINES, SURFACES, AND SOLIDS.

76. It is sometimes more convenient to refer to the traces of a line, surface, or solid, than to its projections; and as we use traces more or less in all our drawings, though they have hitherto seldom received that name, it will be advisable to define the term and give a few examples.

77. **Trace of a Line.**—*The trace of a line is the point of intersection of the line with a plane; if the line does not touch the plane to which it is referred, then the trace is the point in which it would meet it if it were produced.* We speak of the *traces* of a line, surface, or solid, when they are referred to the co-ordinate planes.

If a line be parallel to the vertical plane, but not to the horizontal, it has one trace, on the horizontal; if it be parallel to the horizontal, but not to the vertical, in this case it has only one trace, on the vertical; if parallel to both planes it has no trace. Hence we see the distinction between traces and projections, for we can always obtain *projections* of a line, but not always *traces*. The following example will perhaps explain this distinction:—Suppose we have a shaft which is inclined to a surface, say a wall, but does not meet it, and we wish to indicate the point in which the centre line of the shaft would meet the wall.

Let *ab* and *a'b'*, fig. 79, represent the plan and elevation, respectively, of the shaft, *cd*, *c'd'* its centre line; and *ef* the plan of the wall.

Let the centre line meet the wall, as shown by its projections c and c'; then c' is the ***vertical trace***, and c the plan of that trace.

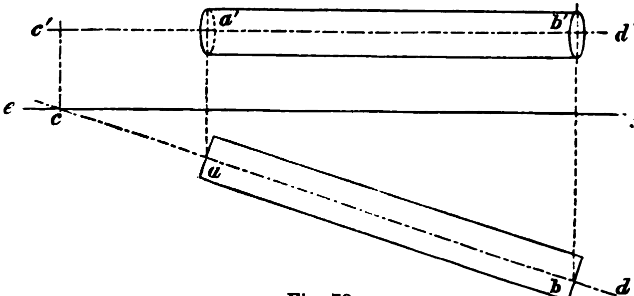


Fig. 79.

Examples of traces will we found in figs. 60, 62, and 76. In fig. 60, h is the horizontal trace of HK; l' the vertical trace of LM; the line NO has no trace, because it is parallel to the co-ordinate planes. In fig. 62, r is the horizontal trace of SR; y' the vertical trace of UT; v the horizontal trace of WV.

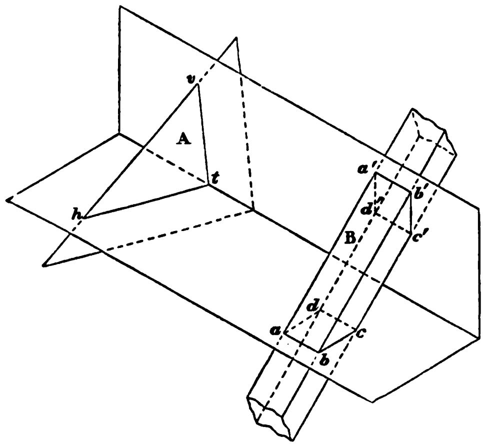


Fig. 80.

In figs. 78, K, L, O, and P, are horizontal traces; T is the vertical, and U the horizontal, trace of RS; its projections are rs' and rs.

78. Trace of a Plane.—*The trace of a plane is the line of intersection of that plane with another plane.*

Let a plane A, fig. 80, meet the co-ordinate planes in the lines ht and vt, then ht is the horizontal, and vt the vertical, trace of the plane A.

79. Trace of a Solid.—*The trace of a solid is the figure formed by the intersections of the surfaces of that solid with a plane.*

The additions made to the traces of a line, Art. 77, apply also to those of planes and solids.

Let a solid B, fig. 80, meet the co-ordinate planes in *abcd* and $a'b'c'd'$, then *abcd* is the horizontal, and $a'b'c'd'$ the vertical, trace of the solid B.

CHAPTER IV.

SECTION I.

GEOMETRICAL SOLIDS HAVING CURVED SURFACES, AND THEIR SECTIONS—THE SPHERE, THE CYLINDER, AND THE CONE.

80. We shall apply the term *solid* to all objects or bodies having volume whether they are solid or hollow.

81. The Sphere.—*A sphere is a solid whose surface is generated by the revolution of a semicircle about its diameter, that diameter remaining fixed in position during the motion.*

Figs. 81 and 82.—These figures represent in two elevations a sphere generated by the semicircle *acb;* the centre *o* of the semicircle is the centre of the sphere, and a radius of the semicircle, as *oc*, is a radius of the sphere. Every projection of the whole sphere is a *great circle*, that is, a circle of the same radius as the generating semicircle.

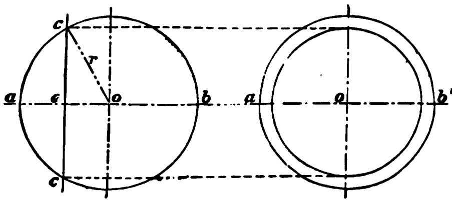


Fig. 81. Fig. 82.

All sections of the sphere are circles; if the section plane passes through the centre, the section is a great circle; if the section plane does not pass through the centre, the section is a *small circle*. In fig. 81,* *cc* is a section plane drawn at right angles to *ab*, the line *cc* is bisected by *ab* at the point *e; ec*

* Figs. 82, 84, 85, 86 are not drawn in section in order that the lettering may appear distinct and plain.

is the radius of the circle which is the section made by the plane *cc*. Fig. 82 is a projection of fig. 81 upon a plane at right angles to the plane of the latter, and parallel to the section plane. The projection of a section of the sphere upon a plane making an acute or an obtuse angle with the section plane, will be considered in the articles on the cylinder.

82. The Cylinder.—*A cylinder is a solid whose surface is generated by a straight line which moves parallel to itself, and always passes through a given curved line; if the curved line is a circle and the generating line is perpendicular to the plane of the circle, the cylinder is a right circular cylinder;* or a right circular cylinder is generated by the revolution of a rectangle about one of its sides, which remains fixed in position during the motion; the fixed side forms the axis of the cylinder.

Fig. 84.

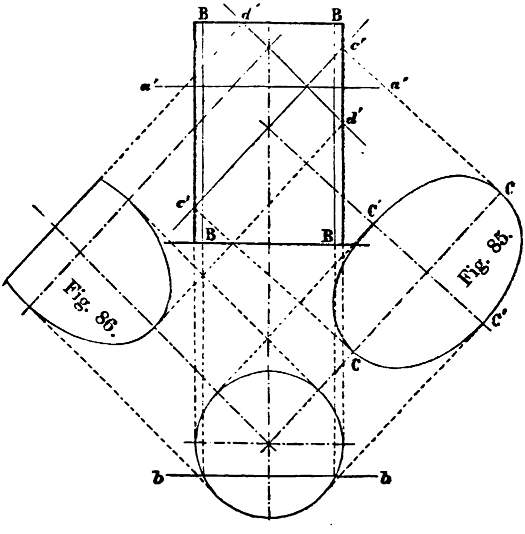


Fig. 83.

Figs. 83 and 84.—These figures represent in plan and elevation a right circular cylinder standing with its base upon a horizontal plane; the plan is a circle, and the elevation a

rectangle. The sides of the rectangle which are parallel to the axis of the cylinder we shall, for brevity, call sides of the cylinder, for we may assume the cylinder to be a prism of an infinite number of sides. There are other projections of the cylinder besides the two named, as oblique projections, and sections; we will enumerate the different sections of the cylinder, which may be classed under four heads, as follows: —Circular, rectangular, elliptical, and a combination of these according to the position of the section plane.

I. If the cylinder is cut by a plane, as *a'a'*, fig. 84, parallel to its base, the section is a *Circle* of the same diameter as the cylinder.

II. If the section plane, as *bb*, fig. 83, is perpendicular to its base, the section is a *Rectangle*, BBBB, fig. 84.

III. If the section plane, as *c'c'*, fig. 84, is inclined to the base and passes through the sides, the section is an *Ellipse*, CC'CC', fig. 85, whose axes are *c'c'*, fig. 84 and the diameter of the circle.

IV. If the section plane, as *d'd'*, fig. 84, is inclined to the base and passes through it or the other end, the section, fig. 86, is made up of a straight line and a portion of an ellipse.

All projections of the cylinder will be included in the sections given above; and as the first two require no further instructions to enable the student to draw them, we will give examples of the third, viz., the ellipse. In Arts. 44...48, pages 42, 43, we have given several methods of drawing an ellipse; there is, however, one which is very useful where construction lines are employed, to which we will now refer.

83. *To determine points in an ellipse, which is the projection of a circle upon a plane inclined to the plane of the circle at an angle other than a right angle.*

Figs. 87...91, Plate II.—The circle, of which the ellipse is a projection, may be any circular section of the sphere, cylinder, or cone. Let A′ 12, fig. 89, represent the plane of the circle, and B 12 the plane upon which the circle is to be projected. Assume the plane of the circle A′ 12 moved to coincide with A 12, at right angles to B 12; describe the circle, fig. 87, and divide it into any convenient number of equal parts, divisible by 4. At a convenient distance below BB draw *bb*, as a centre line for the ellipse; produce the

centre line 6 12, fig. 87, meeting *bb* in *a;* from *a* as a centre describe a circle equal to fig. 87, and divide it into the same number of equal parts. From each point in the circle, fig. 87, draw lines parallel to BB meeting 12 A in points, 1, 2, 3, etc.; from the point 12, fig. 89, as a centre, make 12—1 1—2, etc. in 12 A′ equal to the corresponding distance in 12 A. Draw lines from 12, 1, 2, etc., in 12 A′ at right angles to BB, meeting perpendiculars to these drawn from 12, 1, 2, etc., fig. 88, in the points XII, I, II, etc., fig. 90. Through the points XII, I, II, etc., thus obtained draw the curve, which is the required ellipse. It is not necessary to draw both figs. 87 and 88, either of them may be drawn, and then the required points obtained from the one drawn; we have shown both figures simply to make the construction clearer. Fig. 91 shows a projection upon a plane oblique to both the co-ordinate planes, and is a further example of the method described; the trace of the plane of projection is represented by CD, and the circle remains at the same inclination to the horizontal plane BB as before. As the drawing is similar to the former figure we need only refer the student to the construction lines.

84. The Cone.—*A cone is a solid whose surface is generated by a straight line, which always passes through a fixed point and through the arc of a curve given in magnitude and position; the fixed point is the vertex, and the curve the base; or the cone is a solid, the surface of which is generated by the revolution of a right-angled triangle about one of its sides, which remains fixed in position during the motion, the fixed side forming the axis of the cone.*

Figs. 92 and 93.—These figures represent in plan and elevation a right cone, that is, one whose base is a circle and whose axis is at right angles to that base.

The plan is a circle *afh*, the elevation is an isosceles triangle $f'v'h'$, the base $f'h'$ of which is equal to the diameter of the circle fh, and the lines $v'f'$, $v'h'$ are each equal to a straight line joining the *vertex* or *apex* v' with a point in the circumference of the base. The lines $v'f'$, $v'h'$, are the projections of vf, vh, fig. 92, which, being parallel to the plane of projection of fig. 93, are represented in their true length, while $v'a'$, a projection of va, is considerably shorter than $v'f'$. The length of projections of lines on the surface

of the cone passing through the vertex *v*, and terminating in the circumference of the base, will vary between *v'f'* and *v'a'*. We shall for shortness call the boundary lines *v'f'*, *v'h'* sides of the cone.

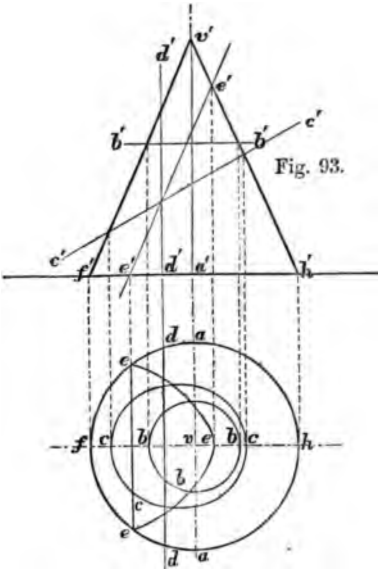


Fig. 93.
Fig. 92.

There are five different kinds of sections of the cone, as follows:—

I. If the section plane, as *a'v'*, contains the axis (and is therefore at right angles to the base in a right cone) the section is an *Isosceles triangle f'v'h'*.

II. If the section plane, as *b'b'*, is parallel to the base, and passes through the sides, the section is a *Circle b b b*.

III. If the section plane, as *c'c'*, passes through opposite sides and makes an angle with the base, the section is an *Ellipse*, of which *c c c* is a projection.

IV. If the section plane, as *e'e'*, is parallel to one side, the section is a *Parabola*, of which *e e e* is a projection.

V. If the section plane, as *d'd'*, is at right angles to the base and does not contain the axis, the section is a *Hyperbola*, fig. 111, Plate VI.

The drawing of sections I. and II. requires no explanation, we shall therefore pass on to the others, and take them in the order given.

85. Figs. 94...96, Plate V.—The Ellipse. Let the circle *a* 0 6, fig. 94, represent the plan, the triangle 0′ *v*′ 6′, fig. 95, the elevation, and *c′ c′* the section plane of a given cone, whose axis is *v′v′* and base *a* 0 6.

Divide the circumference of the circle *a* 0 6 into any con-

venient number of equal parts, divisible by 4, join each of these points to *v*; from 0, 1, 2, etc., fig. 94, draw lines parallel to 3*v′* meeting the base, fig. 95, in 0′, 1′, 2′, etc., and join each of these points to the vertex *v′*. The lines *v*0, *v*1, *v*2, etc., and *v′*0′, *v′*1′, *v′*2′, etc., are the projections of a number of lines on the surface of the cone, each of which is cut by the section plane *c′c′*; the points in which these lines are cut by the section plane are marked 0, I, II, etc., 0...VI, fig. 95, is an elevation, and 0...VI, fig. 94, a plan, of the section; but neither of these shows the true form of section, that is, the form we see when looking at the section in a direction perpendicular to its surface. Fig. 96 shows the required section, which is obtained from figs. 94 and 95; the construction lines will explain how to draw the figure, except perhaps in one or two points to which we will now refer.

In drawing the projection of the section, fig. 94, the distance *v* III is made equal to *e* III, fig. 95, which is a line drawn through III parallel to the base; a circle of a radius *e* III would be the section of a cone made by a plane passing through III and parallel to the base. The line 0...VI, fig. 95, is the major axis. The minor axis BB, fig. 96, can be obtained thus—bisect 0...VI, fig. 95, in *b′*; join the vertex *v′* with *b′* and produce *vb′* to meet the base in *a′*; from *a* draw *a′ a a* parallel to 3′*v′* meeting the plan of the base in *a*, *a*. Join *av*, *av*; make *vb*, *vb*, in *av*, *av*, each equal to one-half *d′d′*, fig. 95, which is drawn through *b′* parallel to 0′ 6′; join *bb*, which is the minor axis. The ellipse, fig. 96, may be drawn as shown by the construction lines, or by one of the constructions given in Arts. 44-48, pages 42, 43.

86. Figs. 97...99, Plate V.—The *Parabola.* Let the circle *ee* 6, fig. 97, represent the plan, the triangle 0′ *v′* 6′, fig. 98, the elevation, and *e′e′* the section plane, which is parallel to 0′ *v′* of a cone whose axis is *v′ v′* and base *ee* 6.

Draw a number of lines on the surface of a cone as in Art. 85, and mark their projections in fig. 97 *v*0, *v*1, *v*2, etc.; and in fig. 98, *v′*0′, *v′*1′, *v′*2′, etc. The plan of the section is marked *ee*...VI...*e*, its elevation *e′*...*e′*, and its true form E...VI...E, fig. 99. The drawing of these figures should present no difficulty as the construction lines show clearly how the points in the curves are obtained.

87. Figs. 97...101, Plate V.—The *Hyperbola.* The section plane is represented by *d d*, figs. 97 and 100, and by *d'* VI, fig. 98. Fig. 100 is a plan of half the cone, the section plane being parallel to the vertical plane upon which fig. 101 is projected.

Lines are drawn upon the surface of the cone as in the previous cases; their projections *v*4, *v*5, *v*6, etc., fig. 100, are cut by the section plane *d d* in points IV, V, VI, etc. The curve D...VI...D, fig. 101, is drawn through the projections of the corresponding points in fig. 100, as shown by the construction lines. The plan of the section is a straight line *dd*, fig. 100; its elevation as seen in fig. 98 is also a straight line; its true form is shown in fig. 101, which is a projection on a vertical plane at right angles to the plane upon which fig. 98 is projected.

88. The student should notice the different methods employed in projecting the sections, figs. 96, 99, and 101. Fig. 96 is obtained from the projections of the points of intersection of the lines on the surface of the cone with the section plane, the elevation being turned into a horizontal plane *sc'*, and then the figure projected from it and fig. 94.

Fig. 99 is obtained in a similar manner, except that the figure is projected from the elevation direct, instead of from the plan, as in fig. 96, the horizontal distances being projected from fig. 97 upon a plane *sp*, which is then turned into the position *sp'*, perpendicular to the section plane *e'e'*, and the figure projected as shown.

Fig. 101 is obtained in the same manner as the two previous figures, except that the plan is turned into such a position that the section is parallel to the plane of projection of the figure; the points in the curve, except D,D, and VI, are projected on to the lines on the surface of the cone from fig. 100.

In drawing such sections as the preceding, the student must use his discretion in selecting which method to adopt, as it will depend upon the relative proportions of the cone, and the position of the section plane, which is best. The sections of the cone may also be obtained by taking a number of horizontal section planes cutting the given section plane.

SECTION II.

Intersections or Penetrations.

89. *When the surfaces of two or more solid bodies are brought into contact, or when one passes through another, the bodies are said to form intersections or penetrations. The line in which the surfaces meet is called the line of intersection.*

Cases of intersection are of constant occurrence in details of machinery, and in other constructions. As the drawing of the intersections of curved surfaces is more difficult than of those having plane surfaces, we shall confine the examples chiefly to the former.

90. Intersection of Cylinders.—The section of a right circular cylinder made by a plane containing its axis is a rectangle whose sides are equal to the length and the diameter of the cylinder. All sections parallel to the section containing the axis are rectangles, having for a common side the length of the cylinder, and for the other side the chord of the circle, cut off by the section plane.

91. *To draw a projection of the line of intersection of two equal cylinders whose axes are at right angles, and are in the same plane.*

Figs. 102 and 103, Plate V.—The projections of the axis are marked *a*, *a′ a′*; *b b*, *b′ b′*; and the plane containing these axes is parallel to the plane of projection of fig. 103. The left-hand half of fig. 103 shows the projection of the line of intersection, which is numbered 0...III...VI; it consists of two straight lines at right angles to each other, and is to be obtained as follows:—

Divide the semi-circumference of the vertical cylinder 3, 0, 3, fig. 102, into any convenient number of equal parts, divisible by 2; and number the points 0, 1, 2, etc. From any point *b′* in the axis *b′ b′*, fig. 103, describe a semicircle 0, 3, 6, and divide its circumference into the same number of equal parts as 3, 0, 3, fig. 102. Through 0, 1, 2, etc., fig. 103, draw lines parallel to *b′ b′*, meeting lines drawn from 0, 1, 2, etc., fig. 102, parallel to *a′ a′*, in points 0, I, II, etc. Join 0...III...VI, which is the required projection of the line of intersection; there will be a similar line on the back half of

the figure. If one of the cylinders is assumed to pass through the other there will be similar lines to 0...VI on the right-hand half of fig. 103.

92. *To draw a projection of the line of intersection of two equal cylinders whose axes contain an angle other than a right angle, but are in the same plane.*

Figs. 102 and 103, Plate V.—The right-hand half of fig. 103 shows the required projection, which is numbered 0...III...VI. The construction lines show clearly how the line is obtained, the method employed being similar to that used in the previous example. It will be unnecessary to work out these two constructions after having mastered the principle employed, as the lines can be drawn at once without any construction lines.

93. *To draw a projection of the line of intersection of two unequal cylinders whose axes are at right angles and are in the same plane.*

Figs. 104 and 105, Plate V.—The projection of the axes of the cylinders are marked *a*, *a′a′*; *bb*, *b′b′*; and the plane containing these axes is parallel to the plane of projection of fig. 105. The right-hand half of fig. 105 shows the required projection, which is numbered 0...VI, and is obtained as follows:—At any point *b*, fig. 104, in the axis *bb* of the smaller cylinder describe a semicircle 3, 0, 3, of a diameter equal to that of the cylinder, and from any point *b′* in the axis *b′b′*, fig. 105, describe a semicircle 0, 3, 6, of the same diameter as 3, 0, 3. Divide the arcs of these equal semicircles into the same number of equal parts and number them as shown. From each point, 0, 1, 2, etc., in the semicircle, fig. 104, draw lines parallel to *bb*, meeting the circumference of the larger cylinder in the points 0, 1, 2, etc.; from these latter points draw lines parallel to the axis *a′a′*, meeting lines drawn from 0, 1, 2, etc., fig. 105, parallel to *b′b′*, in 0, I, II, etc. Join 0...VI, which is the required projection of the line of intersection; there will be a similar curved line on back half of the larger cylinder. If the smaller cylinder passed through the larger one there would be curved lines corresponding to 0...VI on the left hand of fig. 105.

94. *To draw a projection of the line of intersection of two*

unequal cylinders whose axes contain an angle other than a right angle, but are in the same plane.

Figs. 104 and 105, Plate V.—The left-hand half of fig. 105 shows the required projection, which is numbered 0...VI; the method employed to obtain the curve being similar to the preceding example, the student will be enabled to find it without further explanation.

95. The principle employed to obtain the projections of the lines of intersection in Arts. 90-94 can be used for prisms, and, by extending the principle, for other solids. To obtain the line of intersection we have assumed a number of section planes, parallel to the plane containing the axes, passing through the intersecting cylinders; each of these planes will cut the cylinders in points common to both, which will be at the intersection of the lines which form the boundaries of the section. By taking a number of sections we can obtain a corresponding number of points through which to draw the line of intersection. The section planes are represented by 1, 1; 2, 2, etc., fig. 102; and by 1′, 5′; 5′, 1′; 2′, 4′; 4′, 2′, etc., fig. 103.

Again, the intersections of the section planes with the surfaces of the cylinders will be lines on their surfaces parallel to their axes. And the point in which a line on the one surface meets a corresponding line, that is, a line made by the same plane on the other surface, is a point in the line of intersection. Through a number of points thus obtained the line of intersection is to be drawn. We can obtain the same result by drawing a number of lines, generally equidistant from each other, on the intersecting surface, and finding the points where these lines meet the intersecting surface; then through the projections of these points draw the line of intersection, as shown in figs. 104 and 105.

In figs. 102, 103, and 104, 105, the two methods just stated may appear synonymous, and in a sense they are so; but the student will find a greater distinction in the two methods in the case of surfaces which are not cylindrical. The same principles apply to the following example:—

96. *To draw a projection of the line of intersection of two unequal cylinders, the projection of whose axes contains an angle other than a right angle, and is not in the same plane.*

Figs. 106 and 107, Plate VI.—The projections of the axes of the cylinders are marked a, $a'a'$; bb, $b'b'$; and the planes containing these axes are parallel to the plane of projection of fig. 107. In the previous examples the lines of intersection were symmetrical curves, but in the present example the curve is not symmetrical, as will be seen by referring to its projection, which is numbered 0...XI, fig. 107. The back half of the line of intersection is shown by a dotted line; and, as we should expect from the position of the intersecting cylinder, it is not similar to the front half. The curve on the right hand of fig. 107 is similar to the one on the left. As the construction is similar to that employed for figs. 104 and 105, we refer the student to those figures for further explanation if it is required.

97. *To draw a projection of the line of intersection made by a cone passing through the curved surface of a cylinder.*

Figs. 108 and 109, Plate VI.—The projections of the axis of the cylinder, which is vertical, are marked a, $a'a'$, and those of the cone, bb, $b'b'$; the plane containing these axes is parallel to the plane of projection of fig. 109. In this example we have taken the axes in the same plane, but not at right angles to each other; the principal can be applied to the other cases, as may be seen by referring to the intersection of cylinders in the previous articles.

Upon the line 06, fig. 109, which represents the base of the cone, describe a semicircle with that line for its diameter; divide the semicircle into any convenient number of equal parts, say six, and number them as shown. Draw 1, 1; 2, 2; etc., parallel to the axis $b'b'$ and meeting 0, 6 in the points 1, 2, etc.; from these points draw lines to the vertex b'. Draw the plan, fig. 108, of the base of the cone as shown, and from 0, 1, 2, etc., in the base, draw lines to the vertex b; from the points in which these latter lines meet the surface of the cylinder, as represented by the circle in the figure, draw lines parallel to $a'a'$ meeting lines drawn from corresponding points in fig. 109, in 0, I, II, etc. Through these points draw the curve 0...VI., which is a projection of the line of intersection on the left-hand side of the cylinder. The curve on the right-hand side is obtained in a similar manner; and the curves on the back of the cylinder are similar to those

shown, because the axes of the cone and cylinder are in the same plane.

98. Intersections of Cones.—All sections of a right cone made by a plane passing through its vertex and its base are isosceles triangles; the lengths of the projections of the sides of these triangles will vary between $v'f'$ and $v'a'$, fig. 93, p. 71, and the bases between $f'h'$ and naught. Where a cone is the intersecting or intersected body we assume a number of such section planes made to enable us to obtain the line of intersection; in figs. 108 and 109 we have used this method. We may also assume a number of lines to be drawn on the surface of the intersecting cone, as mentioned for the cylinders, and so obtain the line of intersection; in the latter case the lines will represent the lines of intersection between the cone and the section planes previously mentioned.

99. *To draw a projection of the line of intersection made by a rectangular prism passing through a cone.*

Figs. 110 and 111, Plate VI.—The axis of a cone is vertical, and its projections are marked av, $a'v'$; the prism is horizontal, and its ends are rectangular; the axis of the cone is in the centre plane of the prism, and this plane is parallel to the plane of projection of fig. 111.

The intersections made by the vertical surfaces of the prism are portions of hyperbolas (see Arts. 84 and 87, pp. 70, 73); those made by the horizontal surfaces are straight lines in fig. 111, and arcs of circles in fig. 110.

From v, fig. 110, as a centre, describe a circle with diameter mn, equal to the diameter of the cone along $b'c'$, fig. 111, and another of a diameter op; from the points d and k, where these circles meet bc, draw lines parallel to av', meeting $b'c'$ and $g'h'$ in $k'd'$. Join mk', $d'o$, and $d'k'$, the latter is a portion of the hyperbola d'VId', and join similar points on the right-hand side of the figure. Then the required lines are $mk'd'o$, $nk'd'p$; there are similar lines on the back of fig. 111.

100. *To draw a projection of the line of intersection made by a cylinder passing through a cone.*

Figs. 112 and 113, Plate VI.—The axis of the cone is vertical and that of the cylinder horizontal, and both axes are

in a plane which is parallel to the plane of projection of fig. 113. The projections of the axes are marked *av*, *a'v'*; *bb*, *b'b'*. In this example we have obtained the lines of intersection by taking a number of horizontal section planes intersecting the two solids; the sections of the cone made by these planes are circles, and those of the cylinder rectangles; the points in which the circle and rectangle intersect in each plane are points in the intersection.

Describe semicircles from *b* and *b'* as centres, of the same diameter as the cylinder, and divide them as shown; from each of these points draw lines parallel to *bb*, *b'b'*, figs. 112 and 113. From *a* as a centre describe circles of diameters OO, *cc*, *dd*, etc., which will represent sections of the cone made by the horizontal planes; the points where each circle meets the rectangle, fig. 112, made by the same plane, are points in the line of intersection. The projection of the line of intersection in fig. 113 is marked 0...VI., and in fig. 112 by 0...VI...III...0; the right-hand half of fig. 112 shows the whole line of intersection, half of which is dotted. There are lines similar to 0...VI. on the back of fig. 113.

We have now given some of the more common cases of intersection which occur in details of machinery, and have laid down such principles as should enable the student to draw any ordinary case of intersection he may meet with in such details; however, all special cases that occur in the constructions illustrated in this book will be explained.

SECTION III.

DEVELOPMENT OF SURFACES.

101. *If the surface of a solid or hollow body be unfolded so as to represent a plane surface, that surface is called a development or envelope of the body.* Developments are required in many branches of engineering; for example, the *plates* of a boiler when *set-out*, and before they are bent to the required shape, represent the development of the boiler. The apertures for the *dome*, *man-hole*, etc., on the curved surface of a boiler, have to be developed, and it will be obvious that,

if these circles have apertures for their projections, their developments will not be circles. We shall consider the development of some of the simpler cases of curved surfaces, as the right circular cylinder and cone.

102. DEVELOPMENT OF CYLINDERS.—The development of the right circular cylinder is a rectangle having for its sides the length of the cylinder and the circumference of the circle which forms one of its ends; to this should be added two circles equal in diameter to the ends of the cylinder; but in examples that follow we shall simply show the development of the curved surface. The rectangle 3′3′33, fig. 114, Plate V., leaving out the curved lines, represents the development of the vertical cylinder of figs. 102 and 103.

103. *To draw the development of a cylinder having apertures in its surface.*

Fig. 114, Plate V.—This figure represents the development of the vertical cylinder shown in figs. 102 and 103, in which apertures are made by two cylinders equal in diameter to the vertical one. One of these cylinders is at right angles to the axis of the vertical one, the other is not, but their axes lie in the same plane.

Divide the circumference of the circle, fig. 102, into any convenient number of equal parts, divisible by 2, and obtain the lines of intersection 0...VI. Make the line 3...3, fig. 114, equal to the circumference of the cylinder, either by calculation or by setting off small chords of the circle, fig. 102 (see Art. 10, p. 14); divide this line into the same number of equal parts as there are in the circle, and draw perpendiculars to 3...3 from each division. Draw the rectangle 3′3′33, and from 0, I, II, etc., fig. 103, draw lines parallel to 3...3, meeting the perpendiculars from 3...3 in points 0, I, II, etc. Join these points and we have the development of the cylinder, with its apertures as required; the right-hand half of fig. 114 shows the development of the left-hand aperture, fig. 103, and the left-hand half that of the other.

104. In cutting out developments it is often of considerable importance where we make the joint, if that is not decided by other circumstances, since either a better form for *working-up*, or a saving of material, may be effected by considering this point. In fig. 114 we have assumed the surface of the

cylinder to be cut along the line 3′3′, fig. 103, whereas in fig. 115 we have assumed it cut along 0′6′, and so we get a development with the apertures not disposed in the same manner as in the former one.

105. Fig. 116.—Another example of development similar to the previous case is shown in this figure, where the cylinders are of unequal diameter. OIVI is the development of the right-hand half of the vertical cylinder; it will be seen by what precedes that the distances in fig. 116 are taken from 104.

106. *To draw the development of a cylinder having apertures in its surface made by another cylinder passing through it; the axes of the cylinders being in parallel planes.*

Fig. 117, Plate VI.—The position of the cylinders is shown in figs. 106 and 107. Draw the lines of intersection as shown in these figures, then draw *cccc*, fig. 117, the development of the vertical cylinder, and mark off the points in the bottom line, *cc*, corresponding to 0, 1, 2, etc., fig. 106. The construction lines shown should enable the student to complete the figure without further explanation.

107. Development of a Cone.—The development of a right cone is a segment of a circle whose radius is equal to a straight line joining the vertex with a point in the circumference of the base; and the length of the segment is equal to this circumference; to this should be added a circle equal in diameter to the base. The segment 060 v', fig. 120, Plate VI., is a development of the cone shown in figs. 118 and 119.

108. *To draw the development of a right cone.*

Figs. 118-120, Plate VI.—Draw the plan and elevation of the cone, as shown in figs. 118 and 119; from v' as a centre, with a radius $v'6$, describe an arc of a circle, 060, fig. 120, whose length is made equal to the circumference of the base of the cone, by either of the following methods:—

I. Divide the circumference of the base, fig. 118, into any convenient number of equal small arcs, say 12, and step off along 060, fig. 120, a chord of one of these arcs 12 times. Then the whole arc 060 will equal (nearly) the circumference of the base. The smaller the chord taken the nearer will the arc 060 approach its true length; for all ordinary purposes

this method may be employed, as the difference between a chord and an arc of a small segment of a circle may be made very small, and for practical purposes nearly equal to nought.

II. Let the angle $0v'0 = \theta^\circ$, the radius of the base $0v = a$, and the length of a side $0v' = c$. Then we have the following equation connecting these quantities:—

$$\frac{\theta^\circ}{360^\circ} = \frac{a}{c}$$

Example.—Let $a = 1$, $c = 3$; then—

$$\frac{\theta^\circ}{360^\circ} = \tfrac{1}{3}$$
$$\theta = 120^\circ.$$

109. *To draw the development of a frustum of a cone.*

Figs. 118-121, Plate VI.—We will take two examples: for the first, frusta made by the section plane dd, $d'd'$, figs. 118 and 119 (the section is a hyperbola); and for the second those made by the plane eee, $e'e'$ (the section is a parabola).

I. Draw the development of the cone, fig. 120, by one of the methods given in the previous article, and having divided its circumference into the same number of equal parts as there are in the circumference of fig. 118, draw lines from the points 0, 1, 2, etc., to v'. From 1, 2, 3, etc., in the base of the cone, fig. 119, draw lines to v', cutting the plane $d'd'$ in $4'$, $5'$, and VI; from these points draw lines parallel to the base, 0—6, meeting $v'6$ in IV, V, and VI. From v' as a centre, with radii v'IV, v'V, and v'VI, describe arcs of circles cutting the radial lines $v'4$, $v'5$, and $v'6$, fig. 120, in the points IV....IV; through these points draw the curve IV...VI...IV, which is a portion of the required curve. Make 4D, 4D, fig. 120, equal $4d$, $4d$, fig. 118, and join D IV, D IV; then the line D...VI...D is the development of the line on the surface of the cone made by the section plane $d'd'$. The smaller portion of the development cut off by this line is the development of the smaller frustum of the cone.

II. Fig. 121 shows the development of the cone, figs. 118 and 119, upon which are marked the developments of the frusta made by the plane eee, $e'e'$. The drawing of these developments is similar to the previous example, and as the

construction lines show clearly how each point is obtained we need not give any further instructions, but refer the student to the figures, and the explanation given in the example named.

110. In Art. 104, page 80, we referred to the cutting out of developments, and, as a further illustration on this subject, we have shown, in dotted lines, another arrangement of the developments of the frusta of the cone. The line E...VI...E which represents the development of the line on the surface of the cone made by the section plane *eee*, *e′e′*, is the one first drawn; the dotted lines E′VI, E′VI, the second arrangement, and it will be seen they intersect the first line in points, III, III, common to both. We have only shown two dispositions of the developments, but, of course, there are various ways of arranging them. The accuracy of the curved lines in all developments will depend in a great measure upon the number of lines used to determine them, but they should not be so numerous as to cause confusion; they should be more numerous where sudden changes take place in the curves.

CHAPTER V.

Motion—Moving Pieces and their Bearings—Velocity—Ratio—Transmission of Motion—Elementary Combination of Mechanism—Forces—Stress—Strain—Work.

111. We have given in the previous chapters certain principles of drawing which we now propose to apply; but before entering upon that part of our subject, it will be advisable to consider briefly certain matters directly connected with the combined subjects we are now treating, and which can be more easily referred to in a collected form. We shall therefore enumerate certain laws and data respecting the following:—

The kinds of motion—rectilinear, rotary, and helical—of simple moving pieces, and the forms of their bearings.

The transmission of motion, and various means for changing both velocity and direction.

Elementary combinations of mechanism.

The kinds of force and stress.

SECTION I.

THE KINDS OF MOTION—MOVING PIECES AND THEIR BEARINGS—VELOCITY-RATIO.

112. A body moves uniformly or with a uniform velocity when it passes over equal spaces in equal successive intervals of time; the rate at which the body moves is called its *velocity*. The units employed for these quantities are feet and seconds (minutes are sometimes employed). Let s denote the space passed over in feet, t the time in seconds, and v the velocity; then

$$s = vt \ldots\ldots\ldots\ldots\ldots\ldots(1);$$

also $v = \frac{s}{t} \ldots\ldots(2)$, and $t = \frac{s}{v} \ldots\ldots(3)$.

If the moving body does not pass over equal spaces in equal successive intervals of time, the motion is said to be *irregular* or *variable*. If the velocity increases, the motion is said to be *accelerated*, and if the velocity diminishes, to be *retarded*. We may have uniformly accelerated or retarded motion. The direction of motion of a body can be represented by a line which indicates that direction; thus, if we say the body C, fig. 122, moves in the direction AB, we mean it moves from A towards B. In all works on mechanics an arrow is used to denote the direction of motion; the point of the arrow showing the direction.

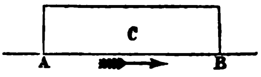


Fig. 122.

When a body moves constantly in the same direction or *path*, it is said to have a *continuous motion;* if it moves alternately backwards and forwards, it is said to have a *reciprocating motion.*

113. Driver and Follower.—When motion is transmitted from one piece to another, either by direct contact, or by means of a connecting piece, the piece whose motion is the cause is called the *driver;* and the piece whose motion is the effect, the *follower.*

The act of giving motion to a piece is termed *driving* it, and that of receiving motion from a piece is termed *following* it.

114. Moving Pieces—Frame of a Machine.—The moving bodies or *pieces* employed in machinery may be divided into two classes: simple or *primary*, and compound or *secondary*. A primary moving piece is one that is directly connected to the *frame* of the machine, and has its motion wholly guided by its connection with it. An ordinary shaft rotating about its axis, or sliding in the direction of its axis, is an example of a primary moving piece; the following are also examples, the piston-rod, the piston, and the slide-block of a steam engine; the loose headstock of a lathe; the table of a planing machine.

A secondary moving piece is one that is not wholly guided by its connection with the frame of the machine. The connecting-rod of a steam engine, and links in general, are examples of secondary moving pieces.

We have used the term *frame* in the sense it is usually employed, that is, to denote the piece or combination of pieces forming the structure which supports directly the primary moving pieces, and indirectly the secondary ones. Examples of this term will be found in many parts of this book.

115. Comparative Motion.—We have often to compare the motion of one moving body or *piece* with that of another, both as regards its direction and velocity; thus, a comparison may be required between the motion of the piston and the slide-valve of a steam engine, or of any two pieces having the same kind of motion. The relation which exists between two moving pieces as to direction is called their *directional relation.* If one of the two connected pieces moves in one direction, and the other piece moves in the same or in any other direction at any given instant, and if they continue to move in those directions, or if, when one changes its direction the other changes its direction also, their directional relation is said to be *constant.*

If one of two connected pieces continues to move in a given direction while the other changes its direction, the directional relation is said to *change.*

The ratio of the velocity of one moving piece to that of another is called their *velocity ratio;* thus let A move with a velocity V, and B with a velocity v, then $\frac{V}{v}$ is their velocity ratio.

Example.—Let A have a velocity of 20 feet per second, B of 10 feet per second, then $\frac{V}{v} = \frac{20}{10} = 2 =$ the velocity ratio of A to B; we need scarcely remark that the kind of motion of each of the moving pieces considered must be the same. The velocity ratio may be constant or varying; examples of both will be given.

In using the term velocity-ratio, it is immaterial whether we take the velocity of the driver or of the follower for the antecedent of the ratio, providing we are consistent and use the one or the other throughout our calculations; we shall take the velocity of the driver for the antecedent.

116. Straight Translation or Rectilinear Motion.—

Straight translation denotes motion in a straight line; it may be continuous or reciprocating; the latter is the form in which it generally occurs in machinery. The piece C, fig. 122, moves in the straight-lined direction AB, and is therefore an example of straight translation; the motions of the piston-rod and the slide-block of a steam engine are further examples; in each of these two cases, the motion is a reciprocating one.

117. Rotation.—Rotation denotes the act of turning about a fixed axis, as for example the motion of shafting generally. In fig. 123, C is a cylindrical piece rotating about the fixed axis AA, whose projections are *aa*, *a'*. Rotation may be either continuous or reciprocating; the latter kind is represented by *rocking shafts*, which are said to oscillate.

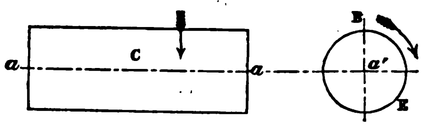


Fig. 123.

It is usual to express the rate of motion of a rotating piece in turns and fractions of a turn per *minute*, in some cases a *second* is taken as the unit of time, a turn being a complete revolution of the piece, thus we say a shaft makes sixty *revolutions* or turns per minute. We may also express the rate of motion in terms of the velocity of the perimeter of the rotating body, as, for example, we may say, a *pulley* has a *perimetral velocity* of 10 feet per second. As these two methods are connected by a very simple relation, we can easily pass from the one to the other.

The number of revolutions in a given time varies directly as the perimetral velocity, and inversely as the radius or diameter of the wheel or pulley.

Example.—A pulley 2 feet in diameter makes sixty revolutions per minute, the velocity of its perimeter in feet per minute $= 60 \times 2 \times \pi$, where π is a constant, and is the ratio of the circumference to the diameter of a circle; therefore, if we reduce or increase the number of revolutions of the pulley we reduce or increase its perimetral velocity

directly in the same ratio. Again, suppose we increase the diameter of the pulley to 4 feet, how many revolutions must the pulley make to have the same perimetral velocity as in the first case? clearly one-half. In the first case the perimetral velocity $= 60 \times 2 \times \pi$, in the second the diameter is doubled, and, therefore, the number of revolutions must be halved to attain the same velocity; thus, $30 \times 4 \times \pi =$ the perimetral velocity, and it will easily be seen that it is equal to the first.

118. ANGULAR VELOCITY.—The rate of motion or speed of a rotating body is sometimes expressed in *angular velocity;* that is, the angle swept through in a *second* by a line, as a radius, in the rotating plane; the angle being expressed in *circular measure.* We may also put it thus: angular velocity expresses how many times the angle turned through by a line in the body, in a second, contains an angle which is subtended by an arc equal to the radius.

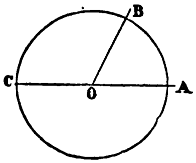


Fig. 124.

In fig. 124 let it be supposed that the radius, OB, has passed from an initial position, OA, and in passing from the one position to the other it has described an arc, AB, equal in length to the radius, OB; the angle AOB is the unit angle. The circular measure of an angle is the ratio which its magnitude bears to the magnitude of the angle AOB.

The circular measure of two right angles $= \pi$ ($= 3{\cdot}1416$), and of four right angle $= 2\pi$.

The number of degrees in the angle AOB $= \frac{180^\circ}{\pi} = 57^\circ{\cdot}29577$.

Perhaps the following way of instituting a comparison between their angular velocities will simplify this matter a little. If two wheels, however they may vary in size, perform a complete revolution in the same time, their angular velocities are identical; but if one wheel turns round *four* times while a second revolves but *once*, the angular velocity of the former is *four* times that of the latter. The angular velocity of the large hand of a clock or watch is twelve times that of the smaller hand. Take the general case where n is

the number of revolutions per minute, and r the radius of the wheel:—

The circumference of the wheel will be $2\pi r$
Since it turns n times per minute;
$\therefore$ Velocity of a point on the circumference per minute is $2\pi rn$;

$$\therefore \text{Velocity per second} = \frac{2\pi rn}{60} = \frac{\pi rn}{30} \text{ this to radius } r.$$

Now angular velocity is calculated on a circle whose radius is unity;

$$\therefore \text{Angular velocity} = \frac{\pi rn}{30r} = \frac{\pi n}{30}.$$

Hence, if 3·1416 be multiplied by the number of revolutions performed by a wheel in one minute, and the product divided by 30, the result will give the angular velocity of one wheel for comparison with another. Now velocity-ratio is independent of actual velocity. Take as an illustration the hands of a clock: the mechanism is so arranged and contrived that, while the hour hand revolves uniformly in twelve hours, the larger hand revolves uniformly twelve times; and if we move the minute hand round once, the hour hand will move only over one-twelfth the space; so that the two hands have always the same relative angular velocity. Every machine has a similar, however, modified arrangement.

119. We can easily pass from *revolutions per minute* to *angular volocity;* thus, for example, suppose a shaft makes ninety revolutions per minute, what is its angular velocity? Multiply the number of revolutions per minute by $\frac{710}{113}$, or 6·2832, and divide by 60; the quotient is the angular velocity; thus the angular velocity $= 90 \times \frac{6 \cdot 2832}{60} = 9 \cdot 4248$. If we wish to convert angular velocity into revolutions per minute we must multiply the angular velocity by $\frac{113}{710}$, or 0·159155, and then by 60, to bring it to minutes.

120. Right and Left-Handed Rotations.—It is usual to distinguish the direction in which a body rotates into *right-handed* and *left-handed;* thus, if the body rotates in the direction of the hands of a watch it is *right-handed*, and *left-handed* if *vice versâ.* If we look at one end of the axis of a rotating body, and then at the other, the direction of rotation is changed; thus, if it is right-handed in the first case it will be left-handed in the second; it is therefore

necessary to refer to one end of the axis only. We have also right-handed and left-handed oscillation, which is distinquished in the same manner as rotation, and right-handed and left-handed screws, where the terms are used in the same relative manner.

121. Velocity-ratio of Rotating Pieces.—As in the case of straight translation, we have often to compare the motion of two rotating pieces both as to their direction and velocity; thus, we have "*directional relation*" and "*velocity-ratio*" for rotating pieces.

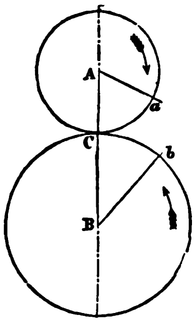


Fig. 125.

The simplest case is that of two circular rotating pieces in contact, as a pair of wheels; if the wheels always remain in contact during rotation their "*directional relation*" will be constant, because, when the direction of motion of one changes, that of the other changes also. In the example shown, if the wheel whose axis is A turns in a right-handed direction, that on the axis B will turn in the opposite direction, as shown by the arrows.

The *mangle* wheel and pinion is an example of "*directional relation*" changing.

122. The *velocity-ratio* of two circular rotating pieces may be expressed in turns or revolutions per minute, or in angular velocity; in each case there is a simple relation existing between the proportions of the two pieces.

Suppose A, fig. 125, is the driver, and it makes thirty revolutions per minute, and the follower, B, makes twenty in the same time, then their radii, or diameters, must be inversely proportional to the number of revolutions; that is, the radius of A : radius of B : : 20 : 30; therefore the velocity ratio of $\frac{A}{B} = \frac{30}{20} = \frac{CB}{AC}$.

Expressed in *angular velocity*, the radii of the two rotating pieces must be inversely proportional to their angular velocities; and conversely, the angles turned through are inversely proportional to the radii.

Let A and B, fig. 125, be the centres of two circular rotating pieces, AB the line joining their centres, C the point of contact; and let the radius $AC = R$, the radius $BC = r$. Suppose a and b on the circumferences of A and B respectively coincided with C before rotation. Then aA, in passing from an initial position CA, has described the angle $a\text{AC} = \text{D}$, and bB, in passing from an initial position CB, has described the angle $b\text{BC} = d$.

$$\text{Then we have } \frac{\text{R}}{r} = \frac{d}{\text{D}}.$$

$$\text{The velocity ratio } \frac{\text{A}}{\text{B}} = \frac{\text{angular velocity of A}}{\text{angular velocity of B}} = \frac{\text{BC}}{\text{AC}} = \frac{r}{\text{R}} = \frac{\text{D}}{d}.$$

$$\text{If R}=2, \text{ and } r=3, \text{ then the velocity ratio } \frac{\text{A}}{\text{B}} = \frac{3}{2}.$$

123. Helical or Screw Motion.—Helical motion is compounded of rotation about a fixed axis, and of straight translation along that axis. This motion is represented by a point, say, on the circumference of a screw; when the screw is turned round the point moves in a helical path or direction, which is made up of the two kinds of motion named.

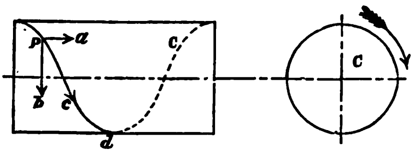


Fig. 126.

In fig. 126, let p be a point on the circumference of the cylinder C; suppose p to have given to it simultaneously a motion of translation along the axis of C, represented by pa, and a motion of rotation represented by pb. Then p would move in the helical path pcd, between the two directions. Helical motion is employed, in the form of *screws*, for transmitting motion, which may be either continuous, as in the case of a *worm* and *worm-wheel*, or reciprocating, as employed for working the *table* of a *planing machine*.

124. Bearings.—By the term *bearings* is to be understood

the *surfaces of contact* between the moving piece and its support; the term *bearing* is usually employed to denote the surface of the supporting piece, still the surface of contact of either piece may be and is called a bearing. Their purpose is to guide the motion of the pieces they support.

The form of the bearing depends upon the kind of motion given to the moving piece; the forms of the bearings for the kinds of motion given in the preceding part of this chapter are as follows:—

I. *Straight Translation.*—The surfaces of the bearings must have a circular, square, triangular, or other straight-lined cross section, and be perfectly straight in the direction of motion; such bearings are called *slides*, examples of which may be seen in lathes, shaping-machines, and some steam engines.

II. *Rotation or Turning.*—The surfaces of the bearings of rotating pieces must be surfaces of revolution accurately turned, as cylinders, spheres, cones, etc. The surface of the moving piece is called a *journal* or *neck*, and the fixed or supporting piece is also called a *journal*, *pedestal* or *pillow-block*, *bush*, *footstep*, *pivot*, etc., according to its position with respect to the supporting frame of the machine. Many examples will be found in the Plates accompanying the work, as Plates XVIII., XIX., XX. These bearings provide also for rocking.

III. *Helical.*—The bearings or swinging motion must have a helical or screw surface. The supporting piece is called a *nut*. For examples see the scraper, screw-cutting lathe, etc. They provide for rotation about a fixed axis and for translation along it.

All bearings must so fit that the intensity of the pressure will not force out the material employed in lubrication.

SECTION II.

THE TRANSMISSION OF MOTION—ELEMENTARY COMBINATIONS OF MECHANISM.

125. THE arrangements employed in the various kinds of machinery for transmitting motion, and for changing it both in magnitude and direction, are so numerous that in the limit of this book we can only give a description of the more common forms of those arrangements. The subject of pure mechanism has been ably treated and reduced to a system by Prof. Willis in his *Principles of Mechanism*, to which we refer those who wish to study the subject.

We shall give a list of certain methods of transmitting motion, and of certain combinations of mechanism; in the chapters that follow, the subjects are treated more in detail.

126. The object of every machine is to perform certain useful work; and to do this it is necessary that it should possess the means of receiving motion from a prime mover, as a steam engine, and of giving out that motion in a form suitable for the operation to be performed. In almost every case the motion received from the prime mover has to be changed either in magnitude or in direction, generally in both, and often in numerous ways in the same machine. Therefore it is clear that in designing a machine for a specific purpose it is necessary to consider what changes of motion are required, and, as there is generally a choice of combinations for producing those changes, it is of the utmost importance that the most suitable arrangement be adopted. To decide which is the most satisfactory arrangement is often a matter of considerable difficulty, especially as practical men may differ very much upon the point in question. Our object will therefore chiefly be to lay before the student such information as will enable him to form his own opinion on the various points that come under his own observation.

127. Transmission of Motion.—When motion is given

to a piece of mechanism, as we have already explained, the act is termed *driving* it, and the act of receiving the motion, *following* it; the piece that communicates its motion to the next is called the *driver*, and the part that receives the motion the *follower*. In fig. 127, if wheel A revolve in the direction of the arrow, and transmit its motion to B, compelling it to move in the opposite direction to the hands of a clock, then A is the *driver* and B the *follower*.

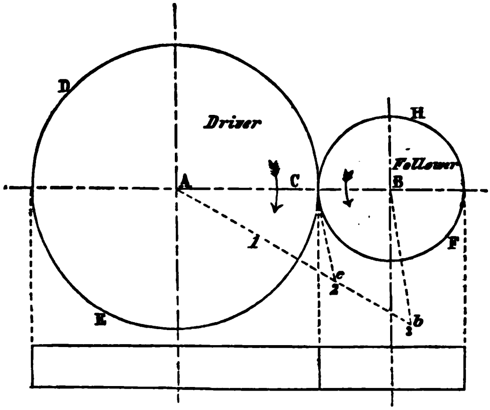


Fig. 127.

Here A and B are the centre of two shafts, let us assume that A is to drive B round twice while it revolves once itself, or B makes two revolutions to A's one; we shall now find the diameters of the wheels. From A draw A*b*, making an angle of about 30° with AB, and upon it set off A*c*, *cb*, so that A*c* = 2*cb*. Join B*b*, and from *c* draw *cb* parallel to B*b*, cutting AB in C, then AC and BC are the radii required. C could have been found by dividing AB by trial; but the method here adopted can be applied whatever the ratio of the diameters of the wheels, being perfectly general in its application.

In fig. 1, Plate IX., where A and B are toothed wheels, if A be the driver and B the follower, then if B is to make two revolutions while A makes one, A must have twice the

number of teeth that B has. It must also be clearly understood that B may be the driver and A the follower. In mechanism the driver and follower do not always move in opposite directions. Motion may be transmitted on four different principles—

(1.) By **Rolling Contact**—as Spur Wheels and Pinions, Crown Wheel and Pinion, Face-Wheel and Lantern, Bevil Wheels, Cones, Rack and Pinion, etc.

(2.) By **Sliding Contact**—as Inclined Plane, Wedge, Cambs, Swash Plate, Crown Wheel Escapement, the Screw, etc.

(3.) By **Wrapping Contact**—as Cords and Pulleys, Belts and Pulleys or Riggers, Speed Pulleys, Capstan, Fusee of a Watch, etc.

(4.) By **Link Work**—as Levers, Cranks, Treadle of a Lathe, etc.

128. We will give a few brief notes and illustrations of each method of transmitting motion as indicated above.

(*a.*) *By Rolling Contact.*—If we consider A and B, fig. 127, to be two smooth wheels or cylinders in close contact, it is very evident that as either moved round its axis it would impart motion to the other. For if they are in contact in any one position they will be in contact in all other positions, and will roll upon each other and impart motion to each other respectively. But there is a practical difficulty in making wheels, under such a condition, revolve uniformly together, hence it is customary to form the edges into teeth; it would at once occur to the most casual observer that the greater the friction the more certainty of motion, and from rough surfaces to teeth is a consequent step.

As another illustration of motion produced by rolling contact, take Plate XVII., where we have a spur wheel and pinion. Suppose A to be the *driver* and B the *follower*, when they act in this manner the term *spur wheel* and *pinion* is properly employed. Pinion A has 15 teeth or

leaves, while the spur wheel has 38, hence their velocity-ratio is as 15 : 38 or $\frac{15}{38}$ or $\frac{38}{15} = 2\frac{8}{15}$, *i.e.*, the driver A must revolve two and a half times and half a tooth more, so that the follower B may revolve once, or if A revolve 38 times B will revolve 15.

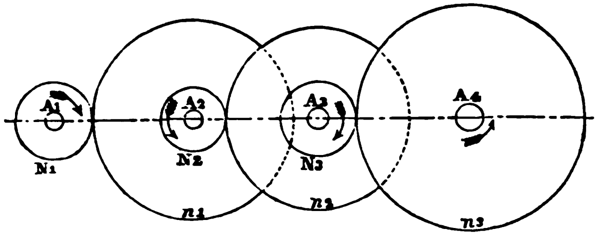


Fig. 128.

Suppose we have such a system of wheels in rolling contact as is figured above. We shall consider axis A_4 or wheel n_3 as the driver, and N_1 as the extreme follower.

Let the diameter of n_3 be 24, of N_3 be 8.
,, n_2 be 18, of N_2 be 6.
,, n_1 be 20, of N_1 be 8.
Now suppose n_3 to make 1 revolution, then N_3 will make 3 ($\frac{24}{8}$).
$\therefore$ when N_3 makes 3 revolutions n_2 will make 3 revolutions.
Now when n_2 makes 1 revolution then N_2 will make 3 ($\frac{18}{6}$).
$\therefore$ when n_2 makes 3 revolutions then N_2 will make 9.
Again when n_1 makes 1 revolution then N_1 will make $2\frac{1}{2}$ ($\frac{20}{8}$).
$\therefore$ when n_1 makes 9 revolutions then N_1 will make $22\frac{1}{2}$.

That is, a train of six wheels, arranged as in our figure, will, upon the shaft A_4 revolving once, cause the shaft A_1 to revolve $22\frac{1}{2}$ times. The general formula will be—

$$\text{Number of revolutions} = \frac{n_3 \times n_2 \times n_1}{N_3 \times N_2 \times N_1}$$

In Chapter VI. this subject is treated more fully.

(*b.*) *By Sliding Contact.*—Bear in mind for a moment the action of a screw, and it is at once apparent that the threads act by sliding contact; in the ordinary vice the screw slides and moves within the female thread; in horizontal engines the parallel motion is obtained through the intervention of a slipper block sliding within guides. We will more parti-

cularly refer to a camb as an illustration of motion obtained by sliding contact.

Fig. 129.

A in fig. 129 is the camb, its purpose is to move the rod B up and down, or give it what is called a vertical reciprocating motion. As the handle is turned so as to move the camb on its centre of motion *c*, in the direction indicated by the arrow, B will be compelled to move up and down. We see that the rod B is at its lowest point *g*, but as the camb moves round, the point *p* comes under the small wheel *a*, at the end of B, when the rod is at its highest point, from which it will fall as the point *p* moves on to its present position. Hence by the camb sliding under the wheel *a* the rod B receives an alternate motion up and down. The construction and use of cambs will be more fully explained.

(*c.*) *By Wrapping Contact.*—Almost all pulleys, cones, and riggers are moved by wrapping contact, *i.e.*, endless bands and cords. The transmission of motion by bands is considered the most simple and inexpensive that is employed; there are many advantages and disadvantages connected with them; they require plenty of room, the shafts may be almost any distance apart, and should any sudden strain be thrown on to the machine, the band will *slip* on the pulley, and thus prevent the destruction of the wheel work. Bands are generally made either flat or round, of leather, gutta percha, hemp, linen, etc.

Figs. 130 and 131 represent the open belt arrangement, figs. 132 and 133 the crossed belt.

In fig. 130, S_1 and S_2 are two parallel shafts, of which S_1 may be considered the driver, the shafts are connected by pulleys and belts, so that for every revolution of S_1, S_2 shall also make a revolution, and the direction in which they turn will be the same, *i.e.*, that shown by the arrows. In

fig. 132 we have the belt *crossed*, which is done when it is required that the pulleys shall turn in opposite directions. It may be noticed that in this second case more belt surface is wrapped round the pulleys than in the first. But we shall again refer to pulleys and bands.

Fig. 130.

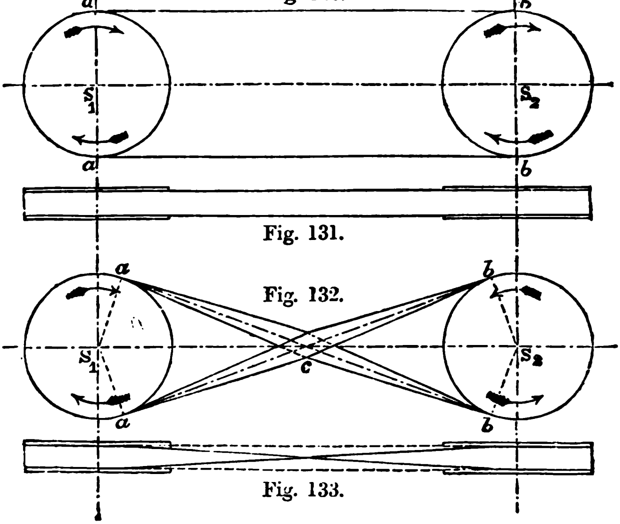


Fig. 131.

Fig. 132.

Fig. 133.

(*d.*) *By Link Work.*—Motion is frequently transmitted by link work, or jointed rods, producing lever motion; the ordinary levers, the treadle of a sewing machine, and the crank of an engine are familiar illustrations. The most general use of such machinery is to convert an ordinary vertical or horizontal reciprocating motion into a circular continuous motion.

129. In the transmission of rotary motion by the pulleys and band arrangement, our choice of kind of motion is very limited; we can, however, connect axes that are parallel, at right angles, or inclined at some other angle, by means of guide pulleys. When the axes are parallel we may use

riggers and cone pulleys, but in all other cases the employment of guide pulleys is essential. Bands and pulleys convey rotary motion from one axis to another; they may also be employed for reversing motion.

Rotary motion may be converted into reciprocating motion in many ways; fig. 129 supplies us with one method, the swash plate is another, the eccentric and eccentric rod also supplies us with a good illustration.

Fig. 134 is intended to represent an eccentric, by which the continuous circular motion of the main shaft is converted into either a horizontal or vertical reciprocating motion.

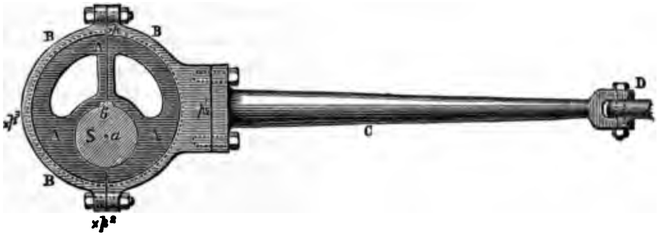


Fig. 134.

S is the main shaft drawn in section; the dotted circle shows a disc or circular plate A completely encircled by a hoop or band B, the disc is free to revolve within the band, to the band B the arm or rod C is attached; the centre line of the rod always passes through the centre of the band or disc *b*; while the extremity D drives a pin connecting it to the slide rod or a bell-crank lever. It should be observed that as the disc A freely revolves within B that the point p will successively assume the positions p^1, p^2, p^3, and hence the end D will be thrown successively backwards and forwards; *a b* or twice *a b* is called the *throw* of the eccentric, where *a* is the centre of the shaft and *b* the centre of the band and disc. By examining this motion closely it will be seen that it is *the camb*, or *the crank* in a modified form.

Many other equally good and appropriate illustrations of the conversion of continuous circular into reciprocating motion might be given, but it is better to refer the student

to a good text-book on Mechanism, such as Tate, Goodeve, or Willis.

Again, it is frequently necessary to convert reciprocating into circular motion. The most familiar instance is in the steam engine, where the reciprocating motion of the piston and piston-rod is converted by the connecting-rod and crank into circular motion at the main shaft.

In the adjoining figure, P is the piston, and PR the piston-rod of a steam engine, the connecting-rod is CR, and the crank CE. As the piston P reciprocates, the point C or crank-pin moves round the dotted circle, the crank CE, being firmly keyed to the main shaft E (shown in section), gives to the main shaft a continuous circular motion; from this main shaft all the required motions of the engine are given off by proper arrangements of mechanism.

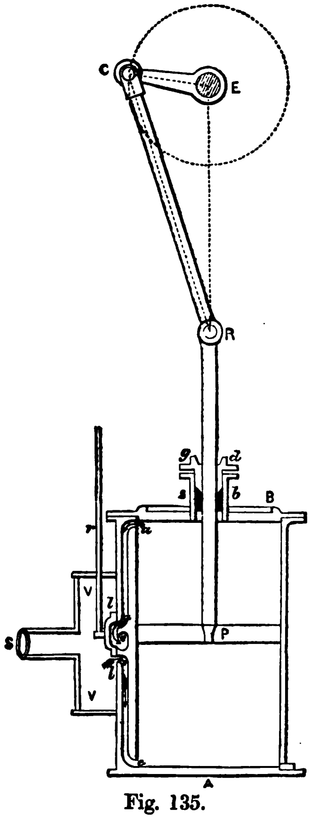


Fig. 135.

130. Ratchet wheels often receive their intermittent circular motion from vibrating pieces. A click or paul is joined to one end of an arm to which a vibrating motion is communicated; as often as the click comes forward it drives one tooth of the ratchet wheel. Feed motions are generally constructed on this principle.

The treadle of a sewing machine, or razor grinders' bench, furnishes an illustration where a reciprocating circular motion is changed into a continuous circular one.

Fig. 136 will exemplify this point; the foot on the treadle AB gives to point B a reciprocating circular motion, which, by means of the connecting-rod BC and the crank CD, communicates a continuous circular motion to the wheel.

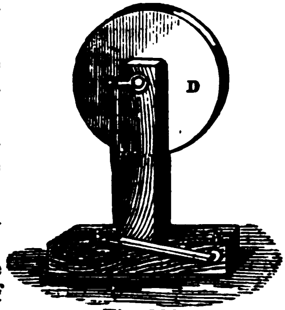


Fig. 136.

These are but a few of the methods of modifying motion, the student must make himself well acquainted with others, it will amply repay the time and labour, for among them will be found the most ingenious contrivances of modern mechanism.

SECTION III.

FORCES—STRESS—STRAIN—WORK—FRICTION.

131. **Force** is that which changes or tends to change the state of a body whether at rest or in motion; or force is that which causes or tends to cause a body at rest to change its position, or that which causes or tends to cause a body in motion to change its rate of motion. In every machine there are certain forces at work, some inducing motion, and others preventing motion taking place. These forces produce or tend to produce certain results; thus a force represented by a load or weight produces stress amongst the particles or atoms of the piece of material to which that load is applied. The load may be sufficient to cause fracture, and break the piece. If the load be not sufficient to cause fracture it may strain it, that is, cause the material to assume some other form to that which it had before the load was applied; under this condition the particles of the material are displaced, and if the strain be very great, the strained piece may be rendered useless for its intended purpose. From this it may be gathered that there may be *stress* in a piece of mechanism and not *strain*. Stress is exerted between bodies or parts of bodies that are contiguous or adherent,

and its intensity is measured in units of weight, the whole weight being proportioned to the area of the surface. We have thrust or pressure, pull or tension, and shear or tangential stress—all different modes in which contiguous particles may act on each other.

132. Work—Horse-Power.—If a man or a steam engine, by the aid of suitable mechanism, raise a weight, or perform some mechanical operation, he is said to perform *work*. It is usual to express the amount of work done in certain units; thus if one pound avoirdupois be raised one foot in the direction of the acting force, one unit of work is said to have been done. And, generally, if m pounds be raised n feet, then mn units of work have been performed, where m and n may represent any numbers whatever, integral or fractional; for it is clear that if one pound raised one foot represents one unit of work, then two pounds raised one foot represent two units of work, and 20 pounds raised 10 feet represent 200 units of work or foot-pounds. In practice it is usual to employ the term *Horse-Power* when we wish to reduce the capabilities of a steam engine to some known standard. *One horse-power* is taken to be represented by *the work done in raising* 33,000 *pounds through the space of one foot in one minute.* The horse-power as thus defined is merely nominal, or a conventional means employed to compare one engine with another. It is as well to observe that this horse-power of 33,000 foot-pounds per minute is sometimes expressed as 550 foot-pounds per second, or 1,980,000 foot-pounds per hour.

Example.—Suppose a steam engine, with the aid of suitable pumps, raises 500 gallons of water per minute from a depth of 150 feet, a gallon of water weighing 10 lbs. What is the horse-power of the engine?—

$$\text{H.-P.} = \frac{500 \times 10 \times 150}{33000} = 22{\cdot}72.$$

133. Modulus of a Machine.—In any given machine that is the best for its purpose which delivers the greatest amount of work in proportion to that applied. There is always a certain amount of work lost in overcoming the inertia of the parts, in friction, and perhaps in imperfect [combi]nation. Were there no loss of work from the causes

now mentioned, the work applied to the receiver would always be equal to that done by the operator; but it is found that the work done by the operator is less than that applied to the receiver; the useful work done is always a fractional part of the work applied, this fraction which has to be determined by experiment or experience for every machine is called the *modulus* of that machine, and tells us at a glance what work is lost. Thus if the work applied to an engine be 40 horse-power, and the operator only delivers 30 horse-power, the modulus is $\frac{30}{40} = \frac{3}{4}$, *i.e.*, one quarter of the work applied to the machine is lost.

134. Moments and Couples.—It is often very convenient to use *moments* in calculations involving stress, strain, etc. The *statical moment* of a force or a pressure is thus defined: The produce of a force into the perpendicular distance of its line of action from a given point is called *the moment of the force with respect to the point, or the moment of the force about the point.*

If an axis be drawn through the point at right angles to the plane which contains the point and the direction of the force, their product is called the moment of the force about the axis.

Let A, fig. 137, represent a given point, EF the line of action of the force F, and p the perpendicular distance of A from EF, then the moment of the force F about the point $A = Fp$. The units employed are feet and pounds. Suppose AB to be a rigid rod which can turn about the fixed point A in the plane of the paper; and suppose a force F to act at B, in the direction EF, then the rod would tend to rotate about the point A. It is usual to distinguish moments as right or left handed, or rather as positive and negative.

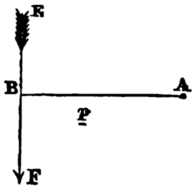


Fig. 137.

Example.—Let $F = 50$ lbs., $p = 2.5$ feet; then the moment about $A = F.p = 50 \times 2{\cdot}5 = 125$ foot-pounds.

Couples.—Two equal parallel and opposite forces acting upon a rigid body, and not acting in the same line, form what is called a couple.

Let F and F represent two equal parallel forces acting in

the directions CE and DH respectively; G the rigid body upon which they act: AB the perpendicular distance between the forces CE and DH, then *these two forces form a couple*. The *arm* or leverage of the couple is the perpendicular distance AB. The *moment* of the couple is the product of one of the forces into the arm; if $AB = p$, then the moment of the couple $= Fp$. The units usually employed are feet and pounds.

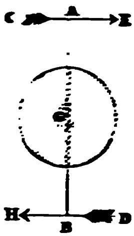


Fig. 138.

The tendency of a couple is to cause the body upon which it acts to rotate, that is, supposing the body to be rigid and free to rotate as a shaft in its bearings. A couple is right or left handed according to the direction in which it tends to rotate. Article 120 illustrates what is meant by right and left handed rotation.

135. Strength.—The strength of a piece of material is ascertained by applying to it a load sufficient to produce strain, fracture, or otherwise, according to the kind of strength we are considering; thus, we have *ultimate strength*, that is the load required to produce fracture; *proof strength*, that is the load which the material ought to sustain without injury to it; *safe load* or *working load* is the load which may be applied with safety to the material, or the load it will sustain with safety; the safe load is always less than the proof load, and of course much less than the load required to cause fracture; it varies in different structures from about $\frac{1}{6}$ in ordinary machines to about $\frac{1}{10}$ in cranes and in wheel work of the ultimate strength.

	Strength.	Strain.	Fracture.
I. Simple or Elementary.	Tensile. Compressive or Crushing.	Extension. Compression.	Tearing. Crushing.
II. Compound.	Shearing. Torsional. Breaking.	Distortion. Twisting. Bending.	Shearing. Wrenching. Breaking across.

Strengths are classified according to the kind of force

acting upon the piece of material or mechanism to produce stress. The forces are *simple* or *elementary*, and *compound;* the foregoing list gives the kind of strength and the corresponding strain and fracture.

136. Tensile Strength.—The tenacity or ultimate strength of a bar or other piece of material is found by applying to it a load acting in the direction of its length, sufficient to cause fracture by tearing it asunder. The bar may be either in a vertical or horizontal position, in the former case the load generally consists of weights applied at the lower extremity, which produce two equal and opposite longitudinal forces; in the second case, these forces must be produced by suitable mechanism, as the hydraulic testing machine. A cord with a weight attached to the end is a simple illustration of a material in tension, the wooden or iron rods of a pumping engine are in tension when lifting water. Tenacity is generally expressed in avoirdupois pounds (or tons) per square inch. The tensile strength varies directly as the area of the cross section of the piece; if the piece vary in its cross section, a section must be taken at the smallest part to judge of the tensile strength. For it cannot be too strongly impressed upon students that the strength of any material, piece, machine, or structure, is the *strength of the weakest part.*

137. Limit of Elasticity.—When a specimen of iron is subjected to a testing machine, it at first extends, and if the strain is not too great, upon being released it returns to its original dimensions; but if the strain exceed a certain limit, the piece of metal receives a "*permanent set,*" and so does not return to its former proportions. This limit is termed *the limit of elasticity*, it varies with every material and with different qualities of the same material. When the limits of elasticity have been determined, the engineer must be careful to subject his structures to strains *very much below* those limits. When a material has once taken a very slight permanent set, it may not be injured, but if the displacement of the particles be great, they never return to the same intimate relationship again, and the material is much damaged. For instance, if a gun be fired with too great a charge of powder, it may cause an alteration of the internal structure,

perhaps invisible on the outside, but yet the overstraining shall be so great that it may be broken to pieces by a hammer, and of course shattered to pieces by the next charge. It has been shown that iron, steel, copper, etc., are exceedingly tenacious. A piece of best steel wire, of given diameter, will support without breaking $7\frac{1}{2}$ miles of its own length. Cables made of fine wire of from $\frac{1}{25}$th to $\frac{1}{30}$th of an inch in diameter possess the extraordinary tenacity of from 60 to 91 tons to the square inch of section.

			Per square inch of section.
Wrought bar-iron has a tenacity of from			25 to 30 tons.
Cast-iron	,,	,,	5 to 15 ,,
Steel	,,	,,	30 to 50 ,,
Copper	,,	,,	$15\frac{1}{8}$,,
Copper wire	,,	,,	$27\frac{1}{8}$,,
Brass (best)	,,	,,	12 to 13 ,,
Ash	,,	,,	7 to $8\frac{1}{2}$,,
Elm	,,	,,	6 ,,
Hornbeam	,,	,,	2 to 9 ,,
Oak (English)	,,	,,	5 to $8\frac{1}{2}$,,
Fir	,,	,,	$3\frac{1}{4}$ to 8 ,,

Wrought-iron girders should never be subjected to sudden strains or violent disturbances from loads exceeding more than one-third the breaking weight. They will bear almost any amount of violent disturbances, so long as the load does not exceed one-fourth the same weight. In a girder properly proportioned, the greatest strain comes upon the bottom section, and a strain on this part of 7 tons per square inch exceeds the limit of safety. The Board of Trade will not allow a railway bridge of wrought-iron to have on it a strain exceeding 5 tons per square inch of section. A bridge will bear with safety 6 tons, or one-fourth the breaking weight, but it is always best to err on the safe side in these matters.

The limit of elasticity in cast-iron is about one-third of its ultimate strength (the ultimate tensile strength of cast-iron is assumed to average 7 tons per square inch); hence it is not safe to strain cast-iron, in tension, above 2 tons to the square inch, for structures exposed to impact, such as railway bridges, where the limiting stress must scarcely exceed a ton.

138. Crushing or Compressive Strength.—The ultimate strength of a piece of material to resist thrust, pressure, or

compression, is ascertained by applying to it a load sufficient to cause fracture by crushing. All columns are subject to compression, in the direction of their lengths; so are walls, bed plates, etc. Wrought-iron resists compression with a force equivalent to 27,000 pounds; *i.e.*, to shorten a piece of iron an inch long, ·533 inches in diameter, the $\frac{3}{1000}$th of an inch requires a force of 27,000 pounds; while the average resistance of cast-iron under the same conditions is 35,000 pounds. The great improvement in modern workmanship permits all parts of a structure that come in contact to be truly faced at the ends, when the uniform bearing thus obtained add greatly to the strength of the whole structure, by enabling it to resist crushing strains most effectually. From experimental researches on iron columns, it has been shown, calling the strength *one* when both ends are flat, that if one end be flat and the other rounded, the strength is diminished *one-third;* if both ends be rounded, the strength is diminished *two-thirds.*

139. Shearing Strength.—If one piece bear upon another so as to act to cause mutual separation, or make one give way by one part sliding over the other, it is said to shear. Consider the boiler plates as riveted together; then if the boiler plate acts so as to cut off the rivets, we have an instance of shearing strain, where each (the plate and the rivet) draws the other sideways in a direction parallel to their surface of contact. Another illustration is supplied by the links of a chain which act upon each other with a shearing stress. The ultimate shearing strain is ascertained by determining the amount of resistance offered by a bar, sheet, rod, etc., of material to the separation of its particles by sliding or shearing. The kind of fracture will depend upon the kind of material subjected to the shearing force.

140. Torsional Strength.—Two equal couples, one right-handed and the other left-handed, and not in the same plane, acting upon a rigid body, as a bar of iron, produce a torsional or twisting strength among the particles of the bar, which tends to cause fracture by wrenching. If the twisting moment is sufficient to cause fracture, then the value of that moment represents the resistance of the particle of the bar to fracture, which is the torsional strength. If the two

couples *ab* and *cd* act upon AC, a shaft or bar of iron in the directions shown by the arrows, they will evidently tend to twist it, and to wrench the particles of the bar asunder. In testing, to ascertain the torsional strength of an object, the testing force is produced by a force acting at the end of a lever attached to a bar or rod, as AB. Experiments are generally made on cylindrical bars 1 inch in diameter, the load being applied to a lever 12 inches long, measured from the centre of the bar; and it has been proved that the average strength of good wrought-iron, under these conditions, is from 700 to 800 pounds, while that of cast-iron is from 650 to 750 pounds, hence wrought-iron is better fitted for shafting, spindles, etc., than cast-iron.

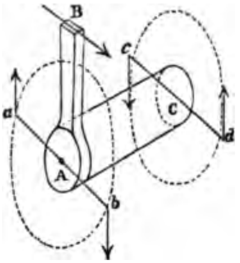


Fig. 139.

Knowing the torsional strength of a one-inch bar, we can readily find that of a two, three, etc., inches; for strengths to resist torsion are as the cubes of the diameters, and inversely.

Example.—Given that the torsional strength of a one-inch bar of steel is 1200 pounds; find that of a three-inch shaft of the same material :—

$$\text{Strength} = 3^2 \times 1200 = 10{,}800 \text{ lbs.}$$

There is a distinction between torsion or breaking strength and *torsional stiffness.* The torsional stiffness of shafting depends upon its length; the total twist is the sum of the twists of each portion; if a shaft be twisted beyond a certain limit, we reach permanent twist, and injury follows; if shafting twists too much, or is not sufficiently stiff, it works with a jerk. "In long shafting it is usual to secure sufficient stiffness by restricting the angle of torsion to some definite limit, say to $\frac{1}{4}°$ per yard length of the shaft, and to secure this amount of stiffness, a larger shaft is often required than would be needed if strength alone were considered. The resistance of shafts of equal stiffness, in this sense, is proportional to the fourth powers of their diameters (*or the square of the area*); that is,

a two-inch shaft will transmit (2^4) 16 times the force which would be transmitted by a one-inch shaft without being twisted through a greater angle."* The working strength of a shaft also depends upon the speed with which it is driven, for evidently the faster it runs, the more easily resistance is overcome.

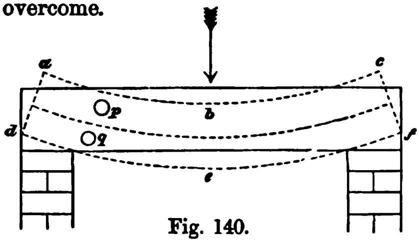


Fig. 140.

141. Bending Strength, or Resistance to Cross Breaking, and Strength of Beams and Girders—*Transverse Strain.*—Fig. 140 represents a beam resting with its ends on two walls. We will suppose the weight acting in the middle in the direction indicated by the arrow. Evidently there will be a tendency in the beam to bend and assume the position indicated by the dotted lines. In this position the upper and lower portions of the beam in section become arcs of two circles; the upper part *abc* is smaller and therefore suffers compression, while the lower *def* is larger, and is therefore under extension. It is clear that there must be some intermediate line between these two where one strain ceases and the other begins, or where there virtually exists no strain. This line is termed the *Neutral Axis*, and is shown dotted between *abc* and *def*. In a rectangular beam of wrought-iron the neutral axis is below the middle of the beam, because wrought-iron presents greater resistance to tension than compression, so the crushing action will extend farther into the beam than the stretching; a beam of cast-iron will have the neutral axis high, because its resistance to compression is great compared to the force with which it resists tension. In pine wood it will be below the middle, since pine resists tension with 5 tons per square inch, and compression with only two. While in oak the neutral axis lies in the middle.

* Anderson's *Strength of Materials.*

From these facts it is evident that a saw cut in the part of a beam under compression does not affect its strength, but one in the under side would weaken it greatly. So also bolts driven in at p will not diminish the strength, while bolts at q and corresponding points will weaken the structure. If we consider the beam, fig. 141, as acted upon by a load at its centre, it will be under the action of three forces, namely, A and B the upward push or resistance of the walls to the weight, and C the weight acting downwards. In what follows the weight of the beam is omitted from consideration.

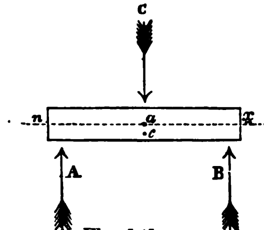


Fig. 141.

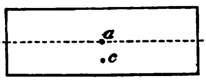


Fig. 142.

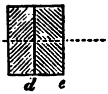


Fig. 143.

It is quite clear that A and B together must equal C, so therefore A is half of C. We must consider the strains as acting round a centre a, a point in the middle of the neutral axis, all the fibres above a being in compression, while those below are under tension. Ba measured parallel to ax will be the leverage which the force B has in trying to crush the beam above the neutral axis nx, and tear it below that line. If we double the length of the beam we double the leverage, and consequently the crushing and stretching forces are doubled, while if we consider the beam with half the length, then these forces will be diminished in the same proportion; *i.e.*, double the length of the beam, it will break with half the weight in the middle, shorten the beam by one-half, it will take twice the load to break it. *Hence the strength of a beam is inversely as its length*, or a beam 10 feet long, other dimensions being equal, is twice as strong as one 20 feet long, three times as strong as one 30 feet long, etc.

By doubling the width of a beam its strength is doubled. If fig. 143 represent the end section of a beam, we may [illegible] d as one beam, and e as a second beam; or the width

of the one beam as doubled. It is very clear that d and e together will be twice as strong as d alone, and that the joint, or its absence, will neither add to or subtract from the strength of the whole, whether the beam be under compression or extension; *that is, the strength of the beam varies directly as its breadth*, double the breadth it is twice as strong, make the breadth three times the original, it is three times as strong, etc.

By doubling the depth of a beam it is rendered 4 (2^2) times as strong, by trebling the depth 9 (3^2) times as strong, etc.

In figs. 141, 142, take a as before the centre of the neutral axis, we have already explained that the strains will act round a as an axis. Let us assume that c is the average leverage of all the fibres below the neutral axis. Now, looking at fig. 142, we perceive at once that there are *twice as many fibres* below the neutral axis as in fig. 141, and that as the beam is twice as wide, their *average leverage is doubled;* therefore their resistance to rupture must be $2 \times 2 = 4$ times that of the fibres in the first beam. The same reason extended to the fibres above the neutral axis would show, that as there are twice as many fibres acting at the end of the leverage to resist compression, therefore the second beam will carry four times the weight of the first, or the strength is quadrupled. *Hence the strength of a beam varies directly as the breadth squared.* From this we see why joists are always put on edge. The conclusions arrived at are: that the strength of a beam supported at both ends, with the weight in the middle, varies *inversely* as its *length, directly* with the *width*, and the *square* of the *depth* or thickness.

Let l=length, d=depth
w=width, C=

a constant depending upon the material employed, which is generally ascertained by experiments, then—

$$\text{Strength} = \frac{w \times d^2}{l} \times \text{C}.$$

If the beam be cylindrical, then $w = d$, and the formula becomes—

$$\text{Strength} = \frac{d^3}{l} \times \text{C}$$

where d diameter corresponds to breadth w and depth.

The following laws hold good with regard to a solid rectangular beam supported at both ends; as regards the amount of deflection or bending that it undergoes when weights are applied for that purpose:—

1. The amount of deflection is proportional to the force applied.
2. The amount of deflection is proportional to the cube of the length.
3. The amount of deflection is inversely proportional to the cube of the depth.
4. The amount of deflection is inversely proportional to the breadth or width.

Let D represent the deflection, Let W represent the weight applied, } also let l = length. d = depth. w = width.

$$D = W . \frac{l^3}{d^3 \times w}.$$

The four laws just stated on the flexure of beams are deduced in precisely the same manner as we have deduced those which govern the strength of beams and girders, and the student must endeavour so to reason them out; of course, they have been fully established by numerous experiments.

Referring again for a moment to fig. 140, p. 109, it will be seen that if the centre of the beam were built into a wall for support, instead of being supported at both ends, one-half of the weight, at present supposed to be at the middle, if applied to the end *cf* or *ad*, would strain the beam exactly as much as at present; or half a beam supported only at one end carries half the weight that the whole beam does when supported at both ends. The whole beam supported at one end will carry $\frac{1}{2}$ of $\frac{1}{2} = \frac{1}{4}$ of what it does when supported at both ends, for we have twice the leverage; hence half the weight will produce the same moment at twice the distance.

When weights are equally distributed over a whole beam supported at both ends, the strain is the same as if half the total weight were placed at the centre; for, supposing each half to act at one-half its former distance, it only acts with one-half the leverage; hence beams and floors support much greater weights when these weights are spread uniformly over the whole surface.

142. Girders. — For consideration we will first take a beam built into a wall with a weight attached to one end; it

will render the matter simpler than if we merely took the weight of the beam acting at its middle point. If our supposed weight were sufficiently heavy to break the beam, the line of rupture would be along *ab*, fig. 144. We will suppose that *de* is the neutral axis, all the material above are in extension, or under a tensile strain, all those below under compression. The reverse of what would happen if the beam were loaded in the middle and supported at both ends. At *c* there is no strain, the forces acting round *c* as their turning point. Take into consideration

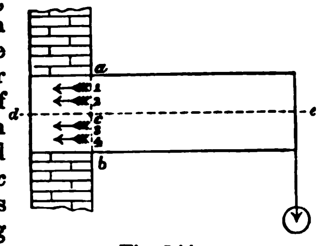


Fig. 144.

a single fibre, say at 1; it will have a leverage of $1-c$ when exerting its force, so also a fibre at 2 has a leverage of $C-2$, but the first has the more leverage, so it will do more work in resisting fracture from tension, while a fibre at *c* does nothing to prevent fracture. Hence the farther the fibres are from *c*, or the wider the beam, the more they assist in preventing rupture and keeping the beam rigid. Applying the same reasoning to the fibres below the neutral axis, we come to the conclusion, that the farther they are from *c*, the greater their force to resist compression and assist in keeping the beam rigid. *Hence, if the materials be taken from the immediate neighbourhood of the neutral axis, and placed beyond a and b, the rigidity of the beam would be increased.*

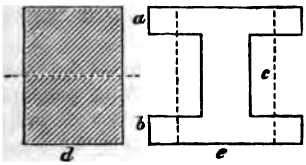


Fig. 145.

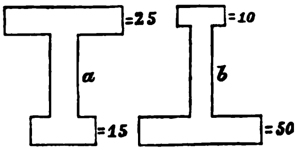


Fig. 146.

In fig. 145 we have in *d* a cross section of a girder to be altered, and *e* shows it when altered. If we suppose that there is the same material in each, it will be found that *e* is

very much stronger than *d*; *e* is what is termed a flanged girder, where *a* and *b* are the flanges and *c* the web. When such a girder rests on its two ends (its general position), and carries a weight, the top flange is under a compressive strain, and the bottom under a tensile strain. Now as these must be always of equal strengths, it becomes necessary to proportion the flanges to the strengths of the materials used under tensile or compressive strain. Taking wrought-iron as resisting 25 tons per square inch of section under tension, but 15 only under compression, then the top flange must be larger than the bottom in proportion of 25:15, or 5:3, as seen at *a*, fig. 146. With cast-iron, the flanges must have the ratio of 10 to 50, or 1:5, as seen at *b*, fig. 146, for cast-iron resists compression more than tension, about five times more. The top flange in wrought-iron is made the larger of the two because the top is under compression, and is weaker under compression than extension; but, on the contrary, as cast-iron is weak under extension, the bottom flange (under extension) is made much larger than the top. Wood is never thrown into these shapes for obvious reasons, such as knots which interfere very greatly with tensile strength, but not with compressive; and the cost of beams altered to such shapes would be greater than those of a rectangular shape capable of carrying the same weight. Occasionally, beams are made in two pieces with the bottom larger than the top.

143. Friction.—Friction is sometimes spoken of as a *modified shearing* stress. It is that force which acts between two bodies at their surfaces of contact to resist sliding movement over each other; this resistance depending upon the weight of the bodies, or on the force with which they are pressed together.

Let us suppose the small block of wood or metal A pressed upon another BC, by means of a force F, acting perpendicular to the two surfaces. This force may be its own weight or a pressure from above. Let us also imagine that a second force, Q, parallel to BC, acts from right to left to move it along the surface BC. Now, since these two forces F and Q are acting at right angles to one another, they cannot counteract each other. So long as the force acting parallel to the surface does not exceed a certain limit, the small block

A does not move, therefore there is some force that we h not yet indicated counteracting Q. This is the force of f tion, which invariably acts parallel to the two surfaces in contact. We may suppose that another force, P, counteracts Q, and prevents it from moving A. Then P will represent the force of friction which is always *for surfaces of the same nature, the same fraction* of the force F applied to press them together, however great the force or large the surfaces in contact. The friction representing the force that prevents motion is called the *co-efficient of friction.*

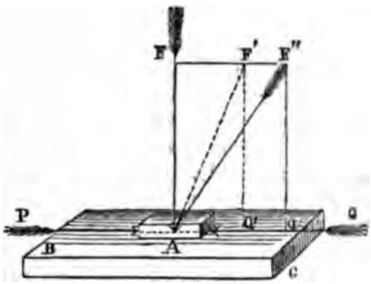


Fig. 147.

Friction in machinery must always be avoided, or great heat will be generated, and damage done. Any amount of heat may be generated by friction, as seen in the brake applied to carriage wheels, railway trains, etc. To prevent friction, axles, journals, bearings, etc., are oiled and greased. Oil keeps a bearing cool, because it lessens the friction; but it must be remembered that no oil will keep a badly turned or an improperly scraped one cool, for the inequalities left by bad workmanship are the best generators of heat.

The action of the lubricant is this: a thin film of the lubricant is partially capable of preventing the surfaces of the two pieces of machinery coming into contact; it thus reduces the resistance due to friction, and assists also in conducting away the heat generated by friction.

The resistance from friction depends not alone on the roughness of the surface, but the force of pressure, the load or work done on the same surface. A double load will produce double the amount of friction, a treble load treble the amount, etc. This statement must be taken within certain limits. Friction does not at all depend upon the magnitude of the surface in contact. Let a block of brass weighing 100 lbs. be placed on a flat smooth surface of cast-iron, it

will require a force of 22 lbs., or $\frac{22}{100} = \frac{11}{50}$ of the whole to draw it along. If another 100 lbs. the same size and shape be attached to the side of the other, it will require 44 lbs. to draw it along, still $\frac{44}{200} = \frac{11}{50}$ of the whole weight. Now let the second block be placed upon the first, so that with the same weight we have only one-half the rubbing surface. Experiments conclusively show that the friction is still $\frac{11}{50}$, or it requires the 44 lbs. still to drag the two weights over the cast-iron, although the surfaces in contact are diminished by one-half. This $\frac{11}{50}$ is called the co-efficient of friction.

The laws of friction received great attention from Coulomb, General Morin, etc. The following are a few of the co-efficients that may possibly prove of service to the engineer. Unguents were not used in their determination, except where expressed.

Oak on oak, when dry,	·48 to ·62.
,, ,, greased,	·07 to ·08.
Wrought-iron on oak,	·49 to ·62.
Cast-iron on oak,	·65.
Wrought-iron on cast,	·19.
Cast-iron on cast,	·16.
Cast-iron axles on lignum vitæ bearings,	·18.
Copper on oak,	·62.
Iron on elm,	·25.
Pear tree on cast-iron,	·44.
Iron axles on lignum vitæ bearings,	·11 (with oil).
Iron axles on brass bearings,	·07 ,,
Wrought-iron on wrought-iron, greased,	·07 to ·08.
Steel on bronze, lubricated,	·1 to ·08.

The two laws of friction may be expressed thus:—(*a*) Within certain limits the friction of any two surfaces increases in proportion to the force applied to press them together. (*b*) The friction is entirely independent of the magnitude of the two surfaces in contact. It must never be forgotten that the friction of motion is wholly independent of the velocity of motion. To reduce friction lubricants are employed, such as grease, tallow, oil, soft soap mixed with oil, black lead, etc., with water and sulphur. The two latter act in a very different manner to the lubricants, and are generally used in extreme cases. The co-efficient of wrought-iron on oak is ·49 in the dry state, but applying water it is reduced to ·26, wh[illegible]p will reduce it to ·21. Oil, tallow, lard, etc., have

all about the same effect, whether it be wood on wood, wood on metal, or metal on metal, the co-efficient being ·07 or ·08, or lying somewhere between; but in the case of tallow interposed between metal and metal the co-efficient rises to ·1. Water reduces the temperature of bearings, because it boils at a very low temperature, and thus a large amount of heat is carried away in steam as latent heat. Sulphur, boiling at a temperature 108° C., acts on the same principle.

Cold water should never be thrown upon a hot axle or bearing, there being a great risk of fracture, owing to the sudden contraction of the metal.

144. Limiting Angle of Resistance, or Sliding Angle.—When a pressure is applied to a body movable upon another fixed body, and both are at rest through the resistance of the two surfaces in contact, but are in a state of equilibrium bordering on motion, the angle at which the one resistance or force inclines to the other is termed the *limiting angle of resistance.* Previously we supposed the force F to act perpendicularly to the surface; let us now suppose it to act at F'' at an angle to BC. Complete the parallelogram $FF''Q''A$ by drawing AF perpendicular to the two surfaces, $F''F$ parallel to the same, and $F''Q''$ parallel to FA. The force F'' may be represented by the line $F''A$, and we may resolve it into the two forces $F''F$ acting parallel to the plane, and FA acting perpendicular to it. Then FA represents the force by which the two surfaces are kept in contact, and the actual friction existing between the two surfaces is a fraction of FA. Suppose we take AQ' as the fraction of AF which represents the friction, and then complete the parallelogram $AQ'F'F$, drawing the diagonal AF'. Now, because AQ' represents the friction on the two surfaces, or that part of AF'' which *opposes the motion* of the block A along BC, and $Q''A$

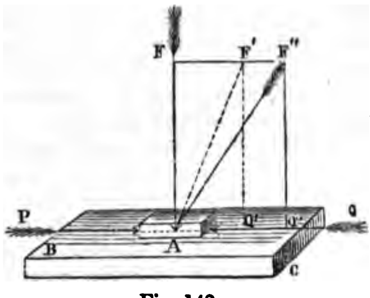


Fig. 148.

represents the forces which F″ exerts to *produce motion* along BC, therefore the body will border on motion when AQ′ = AQ″; it will remain at rest so long as AQ′ is greater than AQ″, and will move when AQ″ is greater than AQ′; or it will move or not according as F″F is greater or less than FF′, or as the angle F″AF is greater or less than the angle F′AF. This last angle F′AF is the limiting angle of resistance. It of course depends upon the co-efficient of friction —the greater the co-efficient of friction the greater the limiting angle. A person walking on the ground and on ice affords a very good illustration of the limiting angle of resistance. As long as the inclination of the legs in walking does not exceed this angle the person does not slip, but the moment it does he slides. The limiting angle of resistance of ice is less than that of the ground, hence we say ice is slippery, and to keep our legs within this angle short steps are taken.

We sometimes hear of *statical friction* and *dynamical friction*. Statical friction is that which opposes the commencement of the motion of one body when in contact with another. Its action ceases as soon as the body moves. We know very well that if one body has been moving over another, and then rests for a time, it is more difficult to move it at first than if we stopped for an instant and then began the motion again. Dynamical friction is always retarding and resisting motion.

Fig. 149 furnishes a very good illustration of several of these points. If the pillar be cut as shown, the resultant pressures must evidently not make with the perpendicular to the planes in contact an angle greater than the limiting angle of resistance; if they did, no resistance of one surface can sustain the force impressed upon it by the other, and they will slip along each other. It is evident that if the cylindrical shaft have a saw cut through *a* parallel to the base, that no force in the direction AB will cause the two pieces to slide, nor is there any force to turn the top piece over on the points *a* or *a′*. We consider

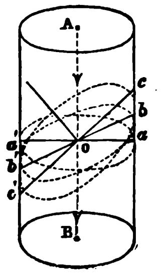


Fig. 149.

the whole as a column with the forces acting through the axis. If several cuts are made, as *b*, *c*, etc., we shall at last arrive at the angle where the limiting angle of resistance is exceeded, and the top slides over the bottom piece. To prevent it slipping, no more is requisite than that the resultant, whose direction is vertical, shall not make with the perpendicular to one of the cutting planes (say bb') an angle greater than the angle of resistance. So long as the plane is not inclined to the horizon at an angle greater than that angle, no slipping will ensue.

CHAPTER VI.

SECTION I.

MATERIALS EMPLOYED IN CONSTRUCTION AND THEIR STRENGTH.

THE drawings in this and the following chapters, with few exceptions, are such as may be considered working drawings, that may be used as examples or drawing copies. The student should not copy off the dimensions by means of dividers, but work from the marked dimensions and proportions given; he should likewise make his drawings to a different scale to that employed in the example. Many of the details in this chapter will be found employed in the examples of modern construction in the last three or four chapters, and the student should try and carefully trace them in the Plates.

145. In this section we shall indicate briefly the chief materials employed in machine construction, together with some of their properties. There is no material so extensively used as iron, for the purposes of machine construction, in its various conditions of purity.

There are two distinct kinds of iron in general use, *wrought* or *malleable* iron, and *cast* iron; besides these two there is an intermediate kind called *malleable cast* iron, which is now being very extensively employed, more especially for light castings. Another condition of iron, the most valuable, and the one that is coming into greater prominence and use every day, is *steel*.

146. Cast-iron is employed in machine construction to a very large extent on account of two important properties it possesses: (1) the ease with which it can be cast into any required form, and (2) its great resistance to compression. There are very many different qualities of cast-iron, all of which possess these peculiar properties in various degrees,

some being more suitable for one purpose and others for another; again, by the mixing of various kinds or brands we can obtain a soft or a hard metal suitable for different purposes. Sir William Fairbairn has shown by experiment that after twelve successive meltings, the maximum of strength, elasticity, and power to resist impact was attained. The maximum of resistance to crushing was attained after fourteen meltings, when it was 95·9 tons per square inch; after the eighteenth it was 88 tons per square inch; after the first re-melting it was 44 tons per square inch.

The average strength to resist a transverse strain of a one-inch bar, $4\frac{1}{2}$ feet between the points of support, has been found to be 471 pounds avoirdupois, whilst its power to resist impact is 817 pounds. Similar iron has an ultimate power of resistance to tensile and crushing strains of $7\frac{3}{10}$ and $41\frac{1}{2}$ tons respectively, or about $1 : 5\frac{1}{2}$.

The strength of a casting depends very much upon its shape and the manner in which it is allowed to cool. In castings, the crystals of which the iron is composed lie in the direction in which the heat passes out of the metal, and as the outward shape and form of the casting determines the rate and direction of cooling, and the points where the heat escapes, we may say that the form and shape of the casting determines its ultimate strength; for it determines the direction in which the crystals lie, "the formation of the long axis being in the line of direction in which the waves of the heat current are passing from the interior of the mass to the nearest point of exit."

The hardness and brittleness of cast-iron are influenced by the rate of cooling. When suddenly cooled—and the same with steel, glass, etc.—the crystals are fixed in a hard, rigid, and brittle state. No time is given them to arrange themselves naturally; hence the iron is very hard, but very brittle. Where the casting is required to be soft, there the mould is made of sand, a slow heat-conducting substance; where it is intended to be hard, the mould is of iron, a quick heat-conducting substance. It is found that cast-iron cooled rapidly is much harder than when cooled slowly, and will resist a greater amount of wear, friction, and compression. Hence chilled castings have, for certain purposes, been much used

lately; by this method of casting, a wheel for a railway carriage has its rim made as hard as hardened steel. We have stated above that there is a difference in the structure of a casting cooled slowly and one cooled rapidly—the one cooled slowly exhibits a large and granular grey structure, while the same cooled rapidly will show a close, white, and brittle structure. A difference in the hardness of different parts of a casting is thus obtained—wherever the bad conductor sand comes in contact with the casting, it will cool slowly, and the parts will be soft; where the good conductor iron comes in contact with the casting, it will cool rapidly, or become *chilled*, and so rendered excessively hard, with rigidity and brittleness. When cast-iron is required to be exceedingly tough, it is cast with what is called a *dead head;* molten metal is allowed to stand on a tube or pipe much above the top of the casting; according to the height of this column of fluid, pressure is conveyed equally in all directions throughout the casting, and hardness and tenacity are produced with a little ductility. A dead head renders a casting solid and its structure close, producing iron fit for hydraulic cylinders. If cast-iron be heated and then suddenly cooled in cold water it is hardened, a white colour is imparted to it, with brittleness and closeness of texture; when once hardened it does not admit of tempering.

147. Malleable Cast-iron.—Cast-iron when it leaves the blast furnace has in it from *two* to *five* or *six* per cent. of carbon, wrought-iron has *no* carbon in it, while steel has from $\frac{1}{2}$ to $1\frac{1}{2}$ per cent. only of carbon; this is the analytical difference between the three. To render cast-iron malleable, more oxygen is added to it; cast-iron is a combination of iron, oxygen, and carbon. To form malleable cast-iron, the iron is subjected to a process of annealing embedded in some substance rich in oxygen; when the latter combines with the carbon in the iron and renders the metal malleable. When steel is made, carbon is added to the pure iron, but here we see the process is that oxygen is added.

148. Wrought or Malleable Iron.—The chemical difference between wrought and other iron is, that it is entirely free from carbon. Its great value is in its great tensile strength, and that it is as strong in the direction of the fibre as it is across the fibre, which is in remarkable contrast

to other materials—wood, for instance, which is very much stronger in the direction of the fibre than across the grain; again, it is malleable, ductile, fibrous, tough, and wealdable, all these properties give to it its great value. Wrought-iron may not be represented with its fibres free and independent of each other, or ready to slide over each other like a bundle of wires; the fibres are firmly bound together side by side with a force very nearly equal to their general tenacity. Cast-iron is purified by refining and puddling, and then by squeezing, rooling, or shingling, is converted into wrought-iron. In the refining and puddling all foreign substances, such as carbon, silicon, phosphorus, sulphur, etc., are eliminated; while the squeezing, rolling, or shingling endow it with its fibrous, malleable, and ductile qualities. During the process of refining or puddling it is the oxygen of the air acting on the impurities of the iron, and combining with them, that renders the iron so pure. The shingling or rolling wrings out from it all mechanical impurities; by the various processes the 7½ tons tensile strength of cast-iron and the 41½ tons compressive strength are altered to 25 tons per square inch ultimate tensile strength, and about 12 tons compressive; it has gained enormously what was wanted. Of course, all iron varies both in tensile and compressive strength.

149. Steel.—The employment of steel in construction has of late years made vast strides; whereas a few years ago it was only used for tools and cutlery, it is now used for railway rails, boilers, bridges, etc. A few years ago its price was enormous; now prices of the best iron and the commonest steel almost touch each other. Considering the slight (chemical) difference between iron and steel, and that steel is but an impure iron, the time must come when steel will be as cheap as iron, and displace iron in all structures requiring strength. When the different processes by which steel is made are considered, it will be readily understood that the difficulties are only mechanical, and knowing what Bessemer has done, advanced thinkers and engineers are looking forward to the time when steel will be commoner than iron. That steel should be produced cheaper than iron is now almost within our grasp.

Steel is essentially a compound of iron and carbon, made

from either cast or wrought iron, by very many different processes; it has been already stated that while cast-iron contains from 2 to 5 or 6 per cent. of carbon, wrought-iron contains no carbon, and steel from $\frac{1}{2}$ to $1\frac{1}{2}$ per cent., rarely exceeding 2 per cent.

To tell a good piece of steel from bad by the mere look is not an easy task, moderately fine grain steel is generally good, or a curved line fracture and uniform grey texture denotes good steel, and the appearance of threads, cracks, or sparkling particles, is a proof of the contrary. A good method to test the quality of steel is to forge a tool and try it. Such a piece need not be wasted, for if it will not do for one purpose it will for another. Steel will not stand so high a heat as iron in forging or welding, the tenacity can be roughly tested by placing a piece on a block of hard cast-iron and crushing it by blows with a hammer; good steel will resist the crushing and cut the hammer face, and bury itself in the cast-iron. Inferior steel, by the same action, will be crushed to powder or beaten flat. As well as resisting crushing power, it has a tensile strength of from 30 to 50 tons per square inch; good steel after plunging into water will require a fair force to break it, and will readily scratch glass. Steel takes a fine polish, but if a drop of nitric acid be applied to it, a black spot is produced; iron remains clean under the same test.

Steel is subjected to tempering, by heating and cooling it can be made to assume almost any stage of hardness. When the chill is rapid it is rendered very hard and brittle. When about to temper a piece of steel, it is heated to redness and cooled in water, and thereby made too strong and hard and brittle; the steel is then again gently heated until it assumes various colours, which are a sure sign of certain stages of hardness, adapting the tool to the purpose required.

150. Copper is the only *red* metal. It is very malleable, and possesses the property of *flowing* in a high degree. It can be beaten out into leaves of extreme thinness, and being very ductile, admits of being drawn into very fine wire exceedingly tenacious—inferior in this respect to iron and steel only. Copper is strongest when cold, and every increment of temperature seems to detract from its strength and tenacity. A cold strip [illegible]ould carry 10,000 pounds was only capable of carrying

7500 pounds when at a temperature of 500° F., and when heated to a red heat in daylight (about 1200° F.) it lost nine-tenths of its strength. From freezing point to boiling point it loses one-twentieth of its strength, at 288° C. it loses one-quarter, at 430° C. it loses one-half. Hence copper is not safe at excessive temperatures. It is used in machine construction for the internal fire-box of locomotives, for brass bronze, bell-metal, gun-metal bearings, vacuum pans, etc. It forms an exception to the general rule of annealing; copper is actually made softer and more flexible by plunging it when red hot into cold water than by any other means. Its tensile strength is about 15 tons per square inch, but in the cast condition it will break with one-half this strain. The remarkable malleability of copper well shows its tenacity; a skilful workman will fashion from a solid block of copper a large vacuum pan for a sugar boiler by well directed and repeated blows, and by carefully annealing the copper after each hammering to set the crystals free again and allow them to flow outwards with more hammering.

151. Brass is an alloy of zinc and copper, one part of zinc to two of copper; but these proportions are varied very much according to the purpose for which the brass is intended. Copper is the chief metal used in alloy with others to form materials for machine construction. Brass is liable to become very brittle when exposed to continual vibration, through a crystalline structure being developed or set up in the metal; if *three* parts of copper be added to two parts of zinc, we obtain a malleable brass, called *Muntz's metal;* occasionally a little lead is added to this latter alloy, which causes it to become more ductile; a large addition would render it brittle. Brass is rather malleable, but will not submit to forging when hot. The tensile strength varies with the alloy, but good brass ranges between 11 and 16 tons per square inch. Its chief use, among others, in machinery is for bushes and bearings.

Bronze is copper and tin—9 of the former to 1 of the latter, is a hard bronze; our bronze coinage consists of 95 of copper, 4 of tin, and 1 of zinc; tin hardens the copper, and it should be remarked in connection with this fact, that the behaviour of bronze is the opposite to that of steel;

bronze heated and quenched in water becomes soft, but heated and cooled slowly, hard.

Gun-Metal is essentially a bronze, $90\frac{1}{2}$ of copper to $9\frac{1}{2}$ of tin, is the general formula; this composition is harder than copper, but more readily melts, and possesses very great tensile strength, from 10 to 18 tons per square inch.

Aluminium Bronze is an alloy of *nine* parts of copper with *one* of aluminium, it resembles gold in colour, but is harder and lighter, very malleable and ductile, and can be forged either cold or hot, but does not weld. It has an average tensile strength of 32 tons, and some specimens have reached as high as 43 tons to the square inch; it is thus seen that it is twice as strong as bronze and nearly equals steel. The price of aluminium prevents this alloy from coming into common use.

152. **Babbit's Metal** consists of a large quantity of tin, a little copper, and twice as much antimony; or about $99\frac{1}{2}$ lbs. of tin, $8\frac{1}{3}$ of antimony, $4\frac{1}{6}$ of copper to every hundredweight of metal. It is very useful for soft metal bearings, reducing the friction very much; it is generally run into an iron caseing or mould, which compels it to retain its shape.

153. **Timber** *used in machine construction.*—Our most useful woods are oak, ash, elm, hornbeam, applewood, holly, pine, lignum vitæ, etc., all of which are more or less used by the enginewright.

For strong, stiff, or durable framework, mahogany, oak, or teak should be used. For tough and pliable framework, ash should be employed. For framework subject to pressure, elm or beech. For levers and connecting rods, strong and tough oak and teak must be used; but if pliability is required, as in carriage shafts, then ash, hickory, or lancewood. For pulley sheaves and rollers, lignum vitæ and boxwood. For bearing for shafts, box, elm, lignum vitæ, or holly, and the end wood should be exposed to rubbing. For cogs, crab tree, hornbeam beech, applewood, and holly.

		Per sq. in.			Per sq. in.	
Ash	resists crushing with	9,000	lbs.,	and has a tenacity	7 to $8\frac{1}{2}$	tons.
Elm	,,	10,300	,,	,,	6	,,
Hornbeam	,,	7,300	,,	,,	2 to 9	,,
Oak, English	,,	8 to 10,000	,,	,,	$3\frac{1}{2}$ to 5 or $8\frac{1}{2}$	,,
Fir	,,	5 to 6 000	,,		$3\frac{1}{4}$ to 8	,,

Ash is used where elasticity and cohesive strength are required, a notable instance being its modern application for springs in agricultural machinery, and its well known and exclusive use for tool handles.

Applewood and *Crab* are valuable as hard woods, being sufficiently hard and durable and fine-grained to resist a great amount of abrasion, hence it is used for cogs in mill work.

Oak is employed because of its durability in water and damp places. Such structures as sluices are built of it, whilst it enters largely into the construction of wooden water-wheels and risers, or buckets for composite ones. Oak forms the best gearing in damp places, for it stands well to its work, where hornbeam or applewood wood "sleep away."

Hornbeam is a white, compact, tough, and hard wood, used for gearing in place of holly and applewood. It does well under heavy strains, but must be kept dry, as exposure to damp and moisture deteriorates it.

Holly is a white, hard, fine-grained, durable wood, and a great favourite where used for cogs, as it possesses toughness with elasticity.

Elm is tough without much elasticity, it stands well in wet places, so it is employed in water-wheels; it is the wood of the wheelwright and boat builder and machinist.

Willow is a durable wood in damp situations, although light and soft. It is tough and wears well in water, almost better than any other wood. It has been employed for the floats of paddle wheels.

Lignum Vitæ is the hardest and heaviest wood known. It is cross-grained, the fibres so crossing each other that it cannot be split with an axe, the peculiar interlacing of the fibres endows it with extreme hardness. The heart of the wood is the part most prized and valued for durability and strength, and is employed for the sheaves of ship's blocks, pulleys, bearings, etc.

Several other kinds of timber are employed, but the above sufficiently indicate the properties that are valuable in machine construction, and will enable the student to judge what is best fitted for his peculiar purpose.

TABLE V.

Tensile and Compressive Strength of Materials.

Material.	Tension in tons per sq. in.	Compression in tons per sq. in.
Cast-iron,	7·3	41·5
Wrought-iron,	25·	12·
Steel,	30 to 50	16 to 34
Copper,	15	26
Brass,	11 to 16	3 to 4
Bronze,	14	6
Aluminum Bronze,	32 to 43	...
Gun-metal,	10 to 18	6
Muntz's metal,	10 to 16	...
Oak,	5 to $8\frac{1}{2}$	$4\frac{1}{2}$
Fir,	$3\frac{1}{4}$ to 8	$2\frac{1}{4}$ to $2\frac{3}{4}$
Ash,	7 to $8\frac{1}{2}$	4
Elm,	6	$4\frac{1}{2}$

These numbers have been taken from the most recent experiments. Authorities differ very much, almost as widely as qualities of metal differ.

SECTION II.

METHODS OF CONNECTING PORTIONS OF MATERIAL.

154. There are two methods more frequently employed than others to connect two or more portions or pieces of material, viz., by *bolts* and *nuts*, and by *rivets;* the former, of which there are numerous forms, is the one most generally employed; however, the method adopted must depend upon circumstances, as the nature of the materials to be connected, etc. Both methods named are employed to form permanent connections; that by bolts and nuts is also employed to form temporary connections.

155. Bolts and Nuts.—Fig. 150 represents an ordinary $1\frac{1}{2}$ inch bolt with hexagonal *head* and hexagonal *nut*, the nut on the left hand, drawn to a scale of $\frac{1}{4}$, is a plan, while the corresponding figures on the right are front elevations, showing the screwed end of the bolt and the nut.

Let D = diameter of bolt and T = thickness of head, then the usual value of $T = \frac{7}{8}$ D. The thickness of the nut is generally made equal to the diameter of the bolt for bolts under 3 inches diameter, nuts for larger sizes of bolts are not made so thick. Nuts are made of three forms—square, hexagonal, and octagonal; the second form is the one chiefly employed. The two faces, through which the bolt passes, are usually *turned*, except for rough work, and one face, as the outside one in the figures, is *chamfered;* there are two common ways of chamfering, that having a *conical outline*, and the one shown in the figures, which has a *spherical outline.*

The table below gives the greatest and least diameters of hexagonal nuts for bolts from $\frac{1}{2}$ inch to 3 inches diameter; these dimensions are taken from a table of *Whitworth Standards*, to which we have made a few additions.

TABLE VI.

A TABLE OF THE SIZES* OF WHITWORTH STANDARD HEXAGONAL NUTS.

Diameter in inches.	Dia. across flats. (*d*)		Dia. across angles (*c*)		Diameter in inches.	Dia. across flats. (*d*)		Dia. across angles (*c*)	
	In Deci-mals.	To nearest $\frac{1}{32}$ of in.	In Deci-mals.	To nearest $\frac{1}{32}$ of in.		In Deci-mals.	To nearest $\frac{1}{32}$ of in.	In Deci-mals.	To nearest $\frac{1}{32}$ of in.
$\frac{1}{2}$	·9191	$\frac{29}{32}$	1·0612	$1\frac{1}{16}$	$1\frac{1}{2}$	2·4134	$2\frac{13}{32}$	2·7867	$2\frac{25}{32}$
$\frac{5}{8}$	1·101	$1\frac{3}{32}$	1·2713	$1\frac{5}{32}$	$1\frac{3}{4}$	2·7578	$2\frac{3}{4}$	3·1844	$3\frac{3}{16}$
$\frac{3}{4}$	1·3012	$1\frac{5}{16}$	1·5024	$1\frac{1}{2}$	2	3·1491	$3\frac{5}{32}$	3·6362	$3\frac{5}{8}$
$\frac{7}{8}$	1·4788	$1\frac{1}{2}$	1·7075	$1\frac{23}{32}$	$2\frac{1}{4}$	3·546	$3\frac{17}{32}$	4·0945	$4\frac{3}{32}$
1	1·6707	$1\frac{11}{16}$	1·9291	$1\frac{15}{16}$	$2\frac{1}{2}$	3·894	$3\frac{29}{32}$	4·4964	$4\frac{1}{2}$
$1\frac{1}{8}$	1·8605	$1\frac{7}{8}$	2·1483	$2\frac{5}{32}$	$2\frac{3}{4}$	4·181	$4\frac{3}{16}$	4·8278	$4\frac{13}{16}$
$1\frac{1}{4}$	2·0483	$2\frac{1}{16}$	2·3651	$2\frac{3}{8}$	3	4·531	$4\frac{17}{32}$	5·2319	$5\frac{7}{32}$

* There are intermediate sizes which are not enumerated in the Table.

There are a variety of forms of bolts for special purposes; thus we have bolts for foundations, as those required for the bed-plate of an horizontal engine, bolts for fixing in stone, bolts where the head is replaced by a *cotter*, etc.

As the cutting of the *thread* in bolts reduces their strength, a saving of material may be effected in the case of long bolts, as foundation bolts, etc., by increasing the diameter of the screwed part *a b*, fig. 150, so that the diameter of the cross section shall not be less than the diameter of a similar cross section of the other part of the bolt.

The forms and proportions of the threads of screws will be considered in a future article.

The material chiefly employed for bolts and nuts is wrought-iron; copper and brass are also used in special cases.

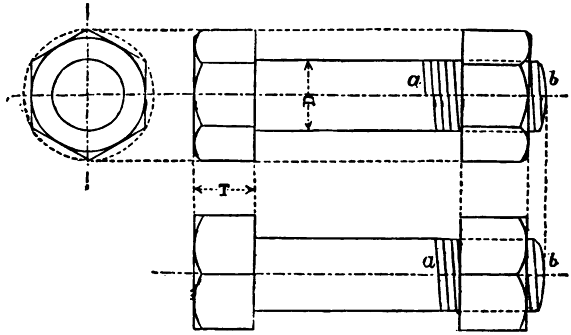


Fig. 150.

156. The drawing of fig. 150 calls for no special remarks, except the hexagonal portions, and as the head and nut are similar, we will take as an example a nut for a 1-inch bolt.

Figs. 35-38, Plate XXIX., are elevations and plans of an ordinary hexagonal nut for a 1-inch bolt, as represented in *scale drawings;* figs. 35 and 36 are not quite correct, but are good approximations. Fig. 36 is drawn first from the given dimensions and then the other figures projected from it. For ordinary scale drawings, sufficient accuracy will be obtained in drawing nuts if we take the following as a rule for nuts

for bolts under $1\frac{1}{2}$ inches diameter:—Make the diameter c, fig. 36, across the angles equal two diameters of the bolt.

The construction lines show how the figures are obtained from fig. 36. The curves $a'b'$, figs. 36 and 37, are represented as arcs of circles, but their true forms are arcs of ellipses, if the outline of the chamfering is spherical, as we shall show in the following figures.

Figs. 38-41, Plate XXIX., represent the nut shown in the preceding figures, drawn full size; the top face is chamfered to a spherical outline of radius r; that is, the inner surface of a hollow hemisphere of radius r would touch the chamfered surface of the nut, a portion of which is represented by $a'\,e'\,f'\,b'$, fig. 39. All sections of the sphere being circles, that made by the vertical plane $v'\,s'$, fig. 38, containing the face $a'\,b'\,c'\,d'$, will be a circle of radius r, which, in the present case, equals one-half the greatest diameter of the nut; the curve $b'\,a'$ is part of this circle. The six faces are all equal, it therefore follows that the curves $a'\,b'$, $b'\,d'$ are also equal; but as those marked $a'\,b'$ are inclined to the vertical plane (at 60° in figs. 38 and 39, and at 30° in figs. 40 and 41), their projections $a'\,b'$, figs. 39 and 41, will not be portions of circles, but of ellipses. The construction lines show how to draw these arcs of ellipses; arcs of circles may be substituted for them, as the error introduced by the substitution is very small.

157. If the nut have a *conical outline*, the curved lines $b'\,a'$ will be portions of hyperbolas; and those marked $a'\,b'$ oblique projections of portions of hyperbolas. As another example we represent in figs. 151 and 152 the nut circumscribed by a hemisphere of radius r in the preceding figures, or we may suppose the nut to be cut out of a solid hemisphere of radius r. In fig. 151 are shown two vertical section planes $v_1\,s_1$ and $v_2\,s_2$, each containing a face of the nut; the plane $v_1\,s_1$ is parallel to the plane of projection of fig. 152, and the plane $v_2\,s_2$ makes an angle of 60° with that plane. On the left of the centre line in fig. 152 is shown one-half of the face contained in the plane $v_1\,s_1$, together with the portion of the sphere cut off by the same plane, which is distinguished by section lines. It is obvious that the upper portion of the

boundary of the face, fig. 151, contained in the plane $v_1\ s_1$, is circular, and so is that of the face contained in the plane $v_2\ s_2$, but as it is oblique to the plane of projection, its projection, shown on the right hand of the centre line, is elliptical. The portion of the section which does not form part of the nut is shown in section.

Fig. 152.

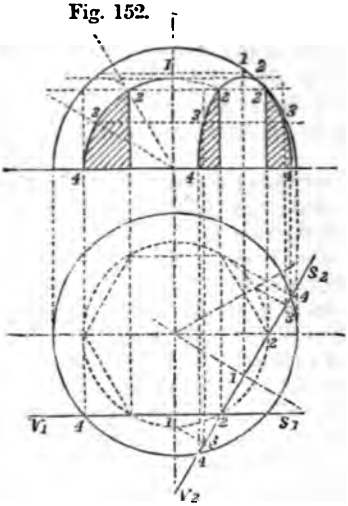


Fig. 151.

158. A common form of bolt and nut connection, usually employed as a permanent connection, is shown in figs. 153 and 154. These figures represent portions of the ends forming the joint of two cast-iron plates connected by bolts and nuts. The whole of the surfaces of the angular portions that form the joint, as seen in fig. 154, are not in contact, but only a narrow strip on each connected piece, which is termed a *chipping-piece*. The bolt has a square head *a*, the portion *b* next to the head is also square, and fits into a square hole, which thus prevents the bolt from turning round while the nut is being *screwed on;* the diagonal lines on *a* and *b* are used to denote the form of their cross section, which is square, but they are as a rule only shown in working drawings. The holes for the bolts in such cases are as a rule *cast in* in the pieces to be connected. Fig. 154 is in section. The figures are drawn to a scale of $\frac{1}{4}$.

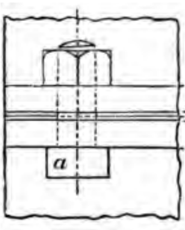


Fig. 153.

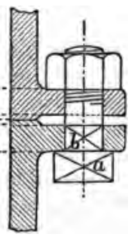


Fig. 154.

159. Figs 155 and 156 represent a common form of bolt and nut connection, employed both as a permanent and as a temporary connection; the bolt is similar to the one shown in fig. 154. Fig. 155 is in section showing the T-headed slot or groove into which the head and part of the bolt next to it fits. Fig. 156 is a plan showing how the bolt is placed in position when the slot does not pass through the lower piece *d*; the slot terminates in a square hole *c*, in the piece *d*, beyond which there is metal. The head of the bolt is first put into the square hole *c*, and then the piece *e*, in which there is a round hole to receive the upper portion of the bolt, is placed in position, the bolt passing through the hole in it; the piece *e* may now be moved into any required position, and the nut, or nut and washer, fixed. In many cases the slot passes through the end of the piece or frame in which it is, as *d*; it is then seen in elevation, as in fig. 155, otherwise the slot would not be seen in that figure, but would be shown in dotted lines. The plan adopted is usually decided by the circumstances of the case.

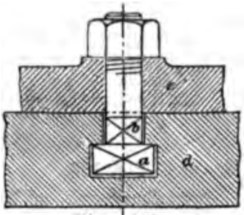


Fig. 155.

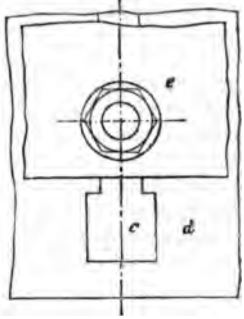


Fig. 156.

160. Figs. 157-159 represent another mode of connecting two pieces by a bolt and nut, chiefly employed for temporary connections. In this example the bolt and its head are cylindrical; the piece *a* can receive a small change of position in the direction *a d*; the bolt is prevented from turning by the *pin* or *key c*. Washers, as *e*, are generally used for such connections, as the one shown in these figures; they are also used in most temporary bolt and nut connections. Fig. 157 is a sectional elevation made by a plane passing through the centre line of fig. 158. Fig. 158 is an end elevation; fig. 159 an end elevation with the bolt in section, showing the form

of the hole in *a* and the pin *c*. If we call fig. 157 a *longitudinl section*, then fig. 159 is a *cross section* of the bolt and pin and part of the piece *b*.

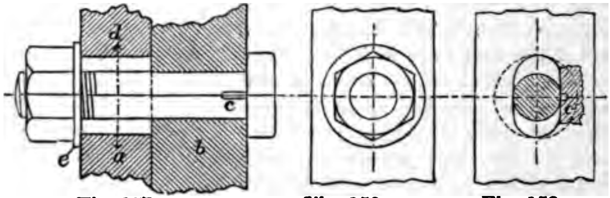


Fig. 157. Fig. 158. Fig. 159.

161. Strength of Bolts.—In calculating the strength of bolts it must be observed that the diameter at the *bottom of the thread* is the smallest diameter of the bolt, and therefore that diameter must be used in our calculations. If we take a cross section through the screwed portion of a bolt, the area of that section is a little greater than that of the bolt at the bottom of the thread, as the thread makes an angle with the section plane; still for safety it is better to keep to the smallest section, as fracture would in all probability take place along that section. Bolts are subjected to two chief kinds of stress—shearing and tensile. The resistance, of the metal usually employed for bolts of wrought-iron, to fracture by these two stresses is about the same. But in the case of shearing, if the bolt does not fit tightly the hole in which it is fixed, the area of its cross section must be increased in the proportion of 4 : 3 for round bolts, for it to be equal in strength to a bolt that does. As bolts have sometimes to withstand sudden strains, and these often when they are also subjected to strains due to overtightening of the nuts, they should have a higher factor of safety than that allowed for the other parts of the machine in which they are employed.

162. Lock-Nuts.—In many machine tools, and in other machines, bolts have often to sustain sudden jerks and vibrations, which tend to loosen the nuts and thereby destroy the efficiency of the arrangement. To obviate this, several arrangements have been made for *locking* the nuts; one common way is by the use of *lock-nuts;* that is, by using two nuts instead of one, the second or outer nut being forced up to the first

one. This arrangement is shown in fig. 160; *a* is the piece of metal through which the bolt *b* passes, *c* is a facing on the piece *a* to receive the nut *d*, and *e* is the lock-nut. This arrangement is a common one for *pedestals* or *plummer-blocks*. Fig. 160 represents one end of the *cap* of a pedestal with the upper portion of the bolt and its nuts.

There seems to exist some difference of opinion among engineers as to the relative thickness of the nuts *d* and *e*. In some cases the nut *e* is made of a less thickness than that of *d*, and in other cases they are of an equal thickness, while in some cases *e* is made thicker than *d*. There is no doubt that the nut *e* acts in all three cases more or less as a lock-nut; and it seems equally clear that the last case is the most efficient one, for the amount of its surface in contact with the bolt exceeds that of the nut *d*. In many cases, where the amount of the vibration is not great, the nut *e* may be equal to, or less than, *d* in thickness, but in all other cases it should be *greater*. The thickness of the nut *d* must be at least one-half the diameter of the bolt at the bottom of the thread; a good proportion would be three-quarters of that diameter for the thickness of *d*, and seven-eighths or once for *e*.

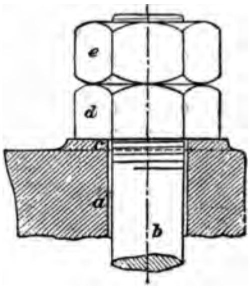


Fig. 160.

163. A method of locking bolts has recently been patented by Messrs. Brown and Bailey, called "Brown and Bailey's Patent Locked Bolt." The screwed end of the bolt has cut upon it a right-handed thread upon which fits the nut proper, as is the case in an ordinary bolt, this nut corresponds to the nut *d*, fig. 160; but instead of this thread going to the end, the diameter of the bolt is reduced for a certain distance from the end, and a left-handed thread cut upon it. Upon this left-handed screw fits the lock-nut, corresponding to the nut *e*, fig. 160. The locking in this case is efficient, but of course there is the extra work involved in the manufacture of the bolt, and as the screws are of different diameters, and one right and the other left handed, the bolts must be made

specially for the purpose for which they are required, as only a very small variation in the thickness of the connected pieces is admissable.

164. Besides the two arrangements of locked bolts described in the previous articles, there are several others in use. In some instances a piece of plate-iron, similar to a double-ended *screw-key*, is made to clip two nuts on different bolts, as the nuts on a pedestal, and thus prevent them from turning. Another method is to fix a set-screw in the nut, and screw it tight against the bolt upon which the nut is screwed; the disadvantage in this case is, that the set-screw is liable to damage the thread on the bolt; this liability may be reduced by interposing between the set-screw and the bolt a piece of soft metal.

Each of the methods enumerated has its special advantage, and in selecting one of them, of course that most suitable for the case under consideration should be chosen.

165. Bolt and Cotter.—It is often necessary to dispense with the heads of bolts, and substitute in their place either a cotter or a screwed end; if the first plan is adopted, as it is in many cases, the arrangement is termed a cottered bolt, and the second plan a stud or stud-bolt. Figs. 161 and 162 represent one form of the bolt and cotter mode of connecting two pieces; *a* and *b* are the two pieces, *a* being connected to *b*; *c* is the bolt, and *d* the cotter. The cotter is employed to hold the bolt in position and to connect it to the piece *b*; the piece *a* is forced up to *b* by the nut *f* on the screwed end of the bolt. In fig. 161 we have shown the cottered end *g* of the bolt

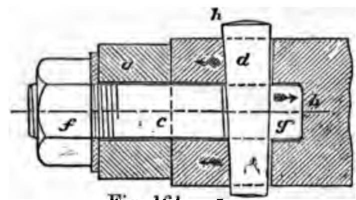


Fig. 161.

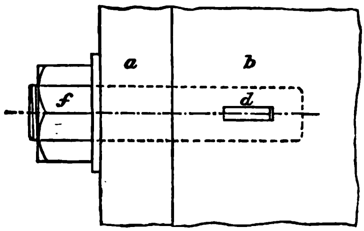


Fig. 162.

contact with the end of the hole in the piece *b*, against ich it is forced by the cotter, the front or left edge of which ırs against the cotter-hole in *b*; and the directions of pressure exerted by the cotter, as it is driven *home*, represented by the arrows. The cotter presses against front edge *h k* of the cotter-hole in *b*, and against the :k edge of the hole in the bolt *c*, clearance being allowed the other edges as shown; the cotter should fit easily on broad sides so as not to wedge tight on those sides. As strength of cotter bolts is reduced by having the cotter-le made in them, it is usual to make that portion contain-; the cotter-hole of a square cross section, and so increase area. If it is desirable to have the cottered ends cylin-cal, then the strength of the bolt may be increased by king the ends through which the cotter passes of a larger meter than the rest of the bolt.

166. Stud or Stud-Bolt.—Studs are employed in a variety cases where ordinary bolts with heads could not be used. some cases they are simply cylindrical pieces of metal with portion of each end screwed; in other cases they have o *flats* made upon the part between the screwed ends, so ıt they may be clipped by a *screw-key* to facilitate the ertion of one end into the screwed or *tapped* hole made receive it. These flats of course tend to weaken the d, and therefore, where strength is essential, it is usual to ke a portion of the stud of a square cross section, the side the square being equal to the diameter of the round part.

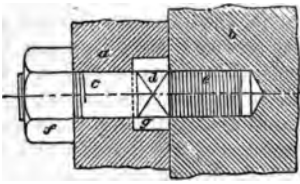


Fig. 163.

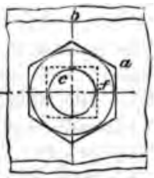


Fig. 164.

In figs. 163, 164, is shown a stud of the form just men-ıed, *a* and *b* are the two pieces to be connected, *a* being smaller; *c* is the stud, and *d* the square portion which is d when screwing the end *e* into the tapped hole in *b*.

The square portion *d* also acts as a collar, so that the stud may be screwed tightly in its place, the collar bearing against the piece *b*. The piece *a* is held in position by the nut *f*; a recess *g* is made in this piece to receive the square portion *d* of the stud. This mode of connection is often employed for the covers of ***steam cylinders***, covers of ***steam chests***, etc. Fig. 163 is a longitudinal section, and fig. 164 an end elevation; they are drawn to a scale of $\frac{1}{2}$.

Examples of the use of studs will be found in the detailed drawings of the steam hammer.

167. In the stud-bolt arrangement described in the previous article, only two pieces are supposed to be connected, but, of course, three or more may be connected, provided they are placed between *d* and the nut *f*; and it is equally clear that as soon as the nut is unscrewed, the whole of the pieces become disconnected.

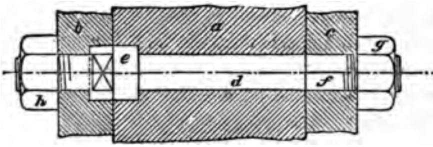


Fig. 165.

Fig. 166.

In figs. 165, 166, is shown an arrangement of stud-bolt by means of which three pieces can be connected, and either of the outer pieces disconnected, without disturbing the other. The three pieces to be connected are *a*, *b*, and *c*; *a* is the middle one and the largest, the bolt *d* has a collar *e* upon it which fits in a recess made in the piece *a*; the end *f* passes through the piece *c*, and has screwed upon it, when in position, a nut *g* which connects the pieces *a* and *e*. On the other side of the collar *e* there is a square portion by means of which the bolt is prevented from turning while the nut *g* is being screwed on; in some cases, instead of a nut, the end *f* may be screwed into another piece of metal, as the piece *c*, which is sometimes necessary. The piece *b* is held in position by the nut *h*.

Fig. 165 is a longitudinal section, and fig. 166 an end elevation with nut *h* and the piece *b* removed; they are drawn to a scale of $\frac{1}{4}$.

168. Set-Screws.—These are bolts used to connect two or more pieces of material where one of the outer pieces acts the part of a nut, that is, the end of the bolt is screwed into the tapped hole of that outer piece, such a bolt is called a set-screw. Set-screws are also used for pressing or forcing one piece of material against another, as shown in fig. 167,

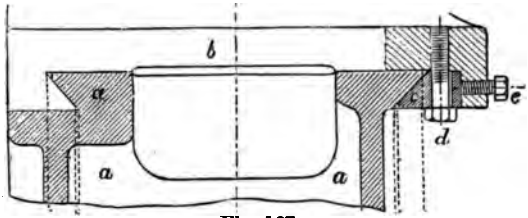


Fig. 167.

where *d* and *e* are two set-screws. The set-screw *e* passes through the piece *b*, which forms a nut and presses against the *strip* *c*, which is in turn forced against the angular surface of *a*. The piece *c* is attached to *b* by the set-screw *d*, which is of the form first described. In some cases the head of the set-screw is *countersunk*. Set-screws are of very common use in all kinds of mechanism.

169. Pins.—Each of the connecting pieces, we have so far considered, is subjected to either tensile stress, shearing stress, compression, or a combination of these. In many connections the stress is simply one of shearing, in which case a pin may be substituted for the bolt; as examples of this kind of connection, we may mention *joints* of *tie rods* and *forked joints* in general.

In figs. 168 and 169 are shown the ends of two rods *a* and *b*, the former has a forked end *c*, into which is fixed the end *d* of the rod *b*, and through the two is placed a pin *e*. The ends *c* and *d* are enlarged, as seen in fig. 169, so that they shall be as strong as the other portions of the rods; generally, the cross sections of the ends are a little greater in area than those of the rods. The pin *e* is round and has a round head *f*, the end *g* has a washer upon it which bears against one facing on the forked end of *a* passing through the pin, and outside this washer there is a *split* pin *g* which

keeps the pin *e* in position. Sometimes the washer is dispensed with, and instead of a split pin an ordinary one of round *wire* is substituted.

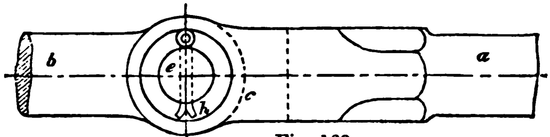


Fig. 169.

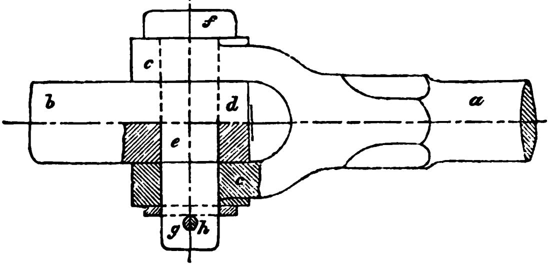


Fig. 168.

In some cases the end *g* is screwed, and one of the holes in the forked end *e* *tapped* to receive it; in this case the small pin, split or otherwise, is dispensed with. Sometimes a bolt and nut is used in place of the pin *e*, but the connecting piece *e* is in all such cases considered to be a *pin*, and is so named.

170. Steady Pins.—Brackets and other fixtures attached to the frame or other portion of a machine, often require to be more firmly connected to that frame than they can be by simply fixing them with bolts and nuts; also, it often happens that a bracket, fixed as stated, has to be removed now and then, and unless some provision were made, it would have to be *reset* each time it was refixed, as the bolt holes, being generally larger than the bolts, are not a sufficient guide in refixing. In such cases *steady pins* are employed; they are simply pieces of round iron or steel, turned, and are driven tight into holes prepared for them in the two connected pieces after the smaller one, as the bracket, is fixed in position. Two such pins at least are, as a rule,

necessary to fix the position of the bracket, and they should be placed as diametrically opposite to each other as possible. In many cases it is convenient to make other provision for the purposes for which steady pins are stated to be used; in such cases they are dispensed with. As a simple and accurate means of fixing brackets, etc., they are very useful.

171. Rivets.—Rivets are cylindrical pieces of metal having generally a head of a hemispherical form; they are used for the purpose of permanently connecting two or more pieces of material, and when fixed in position another head is formed by the process of riveting. It therefore follows that the pieces of material thus connected must be of such a nature as to admit of the riveting process, as in the case of wrought-iron or steel plates for boilers, girders, etc.

Fig. 170 represents an ordinary form of rivet;* allowance is made in the length l for the other head, which is formed when the rivet is put in position. The process of fixing and *heading* rivets is termed *riveting* or *clenching*, and the connection or *joint* is termed a *riveted joint*. The material employed for rivets for wrought-iron and steel structures in general, is a specially prepared tough and ductile wrought-iron called "rivet iron," having a tenacity of about 60,000 lbs. per square inch. Other metals are used for special kinds of work.

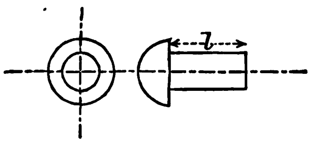


Fig. 170.

In boilers, girders, bridges, and all wrought-iron and steel structures, the rivets are made red hot, put in position, and then the head is formed, either by hand tools or by a riveting machine. We have thus a connecting piece with two heads, corresponding to the bolt head and nut in a bolt and nut connection; but in the present case both heads are permanently fixed. The rivet being put in position when it is hot, draws the connected plates together by its contraction as it cools, thus forming a close and tight joint.

Rivets are subjected to a shearing stress in most cases, and, providing they fit the holes accurately, their resistance

* A table of the dimensions of rivets for boilers is given on p. 144.

to shearing is nearly equal to the tensile strength of the metal of which they are made.

172. Riveted Connections.—There are two methods of forming the joints of riveted connections, one by *butting* or placing the edges of the two plates end to end, forming what is termed a "*butt*" *joint;* by the other method, the plates project or overlap each other, forming a "*lap*" *joint.* In the first example, the butt-joint, it is necessary to employ one or more connecting *cover plates*, which are placed over the joint and riveted to each plate; in the second case, the rivets are put through the two plates where they overlap. In the case of the butt-joint, the plates connected together form one surface; but in the case of the lap-joint, they form two surfaces; hence, in cylindrical boilers that have lap-joints, two adjacent portions, formed by separate rings of plates, are of slightly different diameters.

There are several kinds of riveting in use which differ according to the number and arrangement of the rivets employed. Thus we have single, double-chain or zig-zag, and chain riveting proper; this latter kind was introduced by Sir W. Fairbairn, and is employed chiefly for girder and bridge work. We shall proceed to illustrate some of the various forms of riveted connections, and state the relative strength of each kind of joint.

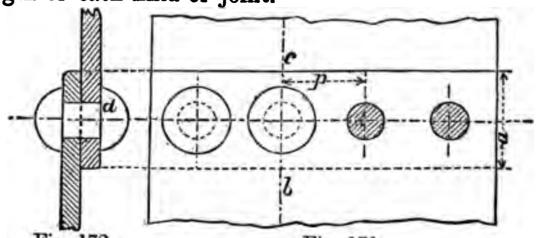


Fig. 172. Fig. 171.

173. Single Riveted Lap-Joint.—In figs. 171 and 172 is shown a single riveted lap-joint, as used for boilers, etc. Fig. 171 is a front elevation, with the rivets on the right of the line *b c* in section; fig. 172 is a cross section through *b c*. The two plates are arranged to overlap each other to a certain extent, and hence the name lap-joint. The *lap* is the

distance a; the distance p of the rivets from centre to centre, is termed the *pitch*, the line containing these centres being midway between the projections of the ends of the plates, as seen in the section. The figures show a portion of the joint of two $\frac{3}{8}$ inch plates, drawn to a scale of $\frac{1}{4}$. The proportion of lap, as used for boilers, for different thicknesses of plate, and the dimensions of the rivets, is given in Table VII.

174. Double Riveted Lap-Joint.—There are two ways of arranging the rivets to form this joint—I. Chain riveting; II. zig-zag riveting; we will take them in order.

I. *Chain riveting.*—In figs. 173 and 174 is shown a chain double riveted lap-joint; fig. 173 is a front elevation, with two rivets in section; fig. 174 is a section made by the plane d e.

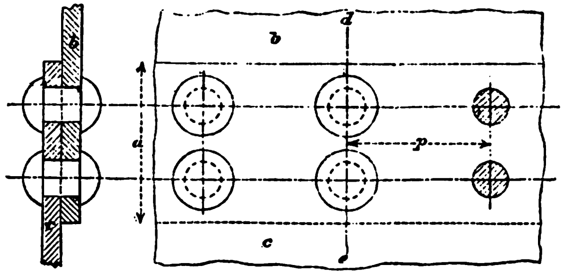


Fig. 174. Fig. 173.

The connected plates are marked b and c, a is the lap. There are two *ranks* of rivets, the centre lines of which are parallel to each other and to the ends of the plates, as shown in the drawing; the distance p, centre to centre of two adjacent rivets, is termed the pitch. In chain riveting, the rivets in each rank are placed behind those in the preceding rank; in the present case there are two ranks, as the joint is double riveted.

II. *Zig-zag riveting.*—In fig. 175 is shown a zig-zag riveted lap-joint. There are two ranks of rivets, the centre lines of which are parallel to each other and to the ends of the plates, but the rivets in one rank are opposite the *spaces* in the other, and hence the term zig-zag. The two connected plates are b and c, the lap a, and the pitch p. The distance between the ranks together with the pitch p determines the angle of the

zig-zag formed by joining the centres of the rivets. We have not given a rule for either of these because of the varying opinions, among boiler-makers, that exist respecting them.

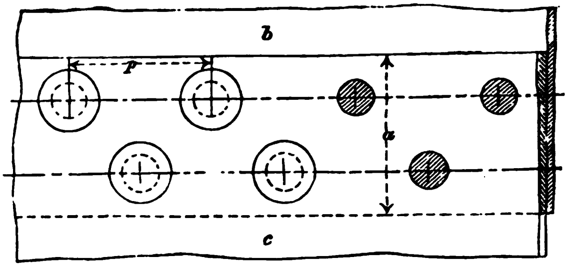


Fig. 175.

The amount of lap as used for boilers for different thicknesses of plate is given in the annexed table.

175. The following table gives the size of rivet, pitch, and breadth of lap for different thicknesses of plates, as used for wrought-iron boilers.

TABLE VII.

DIMENSIONS OF RIVETS, ETC., FOR BOILERS.

The Dimensions are in Inches and Decimals of an Inch.

Thickness of Plate.	Diameter of Rivet.	Length of Rivet from the head.	Pitch.	Breadth of Lap.	
				Single Riveted.	Double Riveted.
·19	·38	·88	1·25	1·25	2·08
·25	·50	1·13	1·50	1·50	2·50
·31	·63	1·38	1·63	1·88	3·13
·38	·75	1·63	1·75	2·00	3·33
·50	·81	2·25	2·00	2·25	3·75
·63	·94	2·75	2·50	2·75	4·58
·75	1·13	3·25	3·00	3·25	5·41

Note.—The pitch is for single riveting; double riveting will be given presently.

176. Single Riveted Butt-joint.—Figs. 176 and 177 represent a single riveted butt-joint with one cover plate; *a* and *b* are the two connected plates, and C is the cover plate. There are two ranks of rivets, one on each side of the joint formed by the plates *a* and *b*, so that each rank connects its plate to the cover plate; the pitch of the rivets is the distance *p*, as before described. If there is only one cover plate, as in the example, it should be at least of the same thickness as the connected plates; if there are two cover plates, one on each side of the plates *a* and *b*, they should be a little more than half the thickness of *a* or *b*. For single riveted joints, the width of the cover plate (or plates) is generally made about $5\frac{1}{2}$ times the diameter of the rivet. In the example, figs. 176 and 177, the plates are $\frac{3}{8}$ inch thick, the rivets $\frac{3}{4}$ inch diameter, and the cover plate $\frac{7}{16}$ inch thick by $4\frac{1}{8}$ inch wide. Fig. 176 is a front elevation, with the rivets on the right of the line *d e* in section, and fig. 177 is a section made by *d e*; they are drawn to a scale of $\frac{1}{4}$.

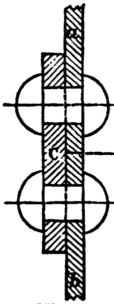


Fig 177.

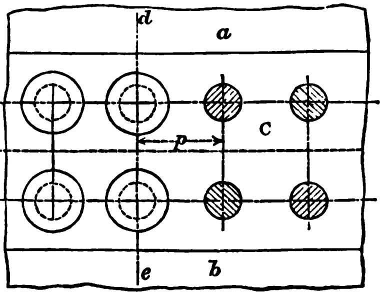


Fig. 176.

In a double riveted butt-joint there are two rows of rivets on each side of the joint, which may be arranged as chain riveting, or as zig-zag riveting, previously described.

177. Chain Riveted Joint.—Where considerable strain has to be borne by portions of built up structures of iron or steel plates, as in the case of the *booms* of *girders* for bridges, etc., it was found that the ordinary double riveted joint was not strong enough, and Sir W. Fairbairn designed what is known as the *chain riveted joint.* This joint is made by simply in-

creasing the amount of the surface of the cover plate or plates, and the number of the rivets, so as to distribute the strain over a much larger area.

Figs. 42 and 43, Plate XXIX., represent a chain riveted joint with two cover plates; there are two thicknesses of plates to be connected, *a* and *b*, and *c*; the plate *c breaks* or covers the joint made by *a* and *b*, which is in the middle of the length of the cover plates *d* and *e*. The riveting consists of 4 *ranks*, *f*, of rivets 4 *files* wide on each side of the joint. The plates are $\frac{3}{4}$ inch thick, the cover plates $\frac{1}{2}$ inch thick, and the rivets 1 inch diameter. The pitch of the rivets is the distance *p*, or 3 inches measured from centre to centre of two adjoining rivets in the same *rank*.

Fig. 43 is a plan showing the cover plate *d*; the rivets on each side of the joint made by the plates *a* and *b* are in section; fig. 42 is half in section; they are drawn to a scale of $\frac{1}{8}$.

178. Strength of Riveted Joints.—It will be obvious that the holes made in the plates at the joints of riveted structures to receive the rivets weaken the plates and, therefore, the joint, by reducing the effective area of the plates; so that in calculating the strength of a riveted plate, the strength at the *joint* must be taken, and not that of the solid plate, because the joint is the weakest part. Fracture of the joint may take place either by tearing the metal between the rivets (along the centre line of the holes), or by detruding the metal between the rivet holes and the edge of the plate. Therefore, in designing a joint, attention must be paid to these two points. Sir W. Fairbairn gives the following rule:—The general rule for proportioning riveted joints is, that the shearing area through the rivets should be equal to the area resisting tearing in the plate after deducting the rivet holes. Let *t* denote the thickness of the plate or plates, *d* the diameter of a rivet, *n* the number of ranks or rows of rivets parallel to the joint, and *p* the pitch in the ranks.

$$\text{Then } p = d + \frac{\cdot 7854\, n\, d^2}{t} \quad \ldots\ldots\ldots (1).$$

For a single riveted joint, $n = 1$, and equation (1) becomes

$$p = d + \frac{\cdot 7854\, d^2}{t}.$$

Example.—We will apply this rule to ascertain what should be the pitch in the case of the single riveted joint shown in figs. 171 and 172.

Here $t = \frac{3}{8}$ inch $= \cdot 38$ inch, $d = \frac{3}{4}$ inch $= \cdot 75$ inch, and $n = 1$;

$$\text{then } p = d + \frac{\cdot 7854\, d^2}{t};$$

$$= \cdot 75 + 1 \cdot 16 = 1 \cdot 91 \text{ inch.}$$

According to Table VII. the pitch should be 1·75 inch, which is a little under that obtained by the foregoing rule.

179. The relative strength of single and double riveted joints, designed as described in the previous article, the joints being of equal widths, is as follows:—

Assuming the strength of the plate to be	100
The strength of a double-riveted joint will be	70
And that of a single-riveted joint	56

Therefore the ratio of the strength of a double riveted joint to that of a single riveted one is as 5 : 4.

Putting the foregoing proportions in another form, and assuming the tensile strength of the iron to be 50,000 lbs. per square inch, we have the strength of the joints per square inch of plate as follows:—

Strength of plate, ...	50,000 lbs.
,, double riveted joint, ...	35,000 ,,
,, single riveted joint, ...	28,000 ,,

These data are the results of experiments made by Sir W. Fairbairn.

180. The strength of a well-designed chain riveted joint, with a cover plate on each side of the jointed plates, may approach very nearly to that of the plate itself; this was shown by an experiment made by Sir W. Fairbairn. Such a result may be exceptional; however, the strength of such a joint generally approaches to 85 per cent. of the strength of the plate.

The strength of a riveted joint also depends on the mode of forming the rivet holes, whether *punched* or *drilled*. It often happens, in the case of punched holes, that, when the plates to be connected are brought together, the holes are not concentric, and then to make them so they are *drifted*, which of course tends to weaken the plates still more, by ill usage, and by reducing the area of the plate. The damage caused by drifting the holes may be avoided by using a cut-

ting instrument, as a reamer; still there is the loss of area due to the enlargement of the holes, and so the joint is weakened even by this process. Another defect arising from the rivet holes in the connected plates not being concentric is, that the rivets do not always fill the holes, and the shearing strength of the rivets is thus reduced.

Recent experiments have proved that joints having drilled rivet holes are much stronger than when the holes are formed by punching, the gain in strength being about 10 per cent. This arises from the fact, that drilling does not damage the plates as punching does, and also that the *pitching* of the holes is generally more perfect in drilled work than in punched, and therefore the rivets fill the holes and make a more perfect connection.

181. Since writing the foregoing, further particulars respecting the relative value of punched and drilled rivet holes have been published, which we give below. No doubt, in time, even more satisfactory data will be furnished, either by Mr. Kirkaldy, or through some other source; and that, in addition, the whole question of riveted joints will be investigated, so as to bring our knowledge of these connections on a level with the present state of exact workmanship: most of the data we possess is rather old, and refers to punched work chiefly.

Some of Mr. Kirkaldy's latest tests had given the following results,* the comparison is made between plates similar in every respect, except that the one has been punched and the other drilled :—

TABLE VIII.

Wrought-Iron Plates.	Punched Holes.	Drilled Holes.
	Per cent.	*Per cent.*
Reduction of strength of net section between holes as compared with equal area of solid plate,	8·7	0·0
Elongation of holes at rupture,	7·65	15·8
	Lbs.	Lbs.
Ultimate tenacity along fibre,	23,292	27,982
,, ,, across fibre,	22,063	25,600

* *Engineering*, May 5, 1876.

182. Examples of Riveted Structures.—In figs. 44 and 45, Plate XXIX., are shown a portion of a wrought-iron plate girder, illustrating how the several pieces are connected together by rivets. The girder consists of a central plate *a*, called the *web-plate*, made up of separate plates; if the girder is of such a length that one plate cannot be conveniently obtained, this remark also applies to the other plates used; top and bottom plates *b*, *b* forming the *flanges* or *booms;* and lengths of angle-iron *c*, *c*, *c*, *c*, which are riveted to the web and the booms, the whole forming an H-shaped cross section, as seen in fig. 44. The joints in the booms have cover plates *d*, *d* over them; the rivets in these places pass through plate or plates, cover plate, and angle-iron. At intervals in the length of the girder are fixed plates *f*...*f*, on each side of which are pieces of angle-iron *e*...*e*, bent to the form shown in fig. 44; these pieces of angle-iron are riveted to the plates *f*...*f*, the web *a*, and the booms *b*, *b*. These plates, with their angle-iron connecting pieces, form stiffening or strut plates, and add to the strength of the girder. At each end of the girder there is an end plate *g*, which is connected to the web and booms by angle-iron.

The top boom in fig. 45, which is a longitudinal elevation of a portion of the girder, is in section made by the plane *h k*, fig. 44, which is a cross section, with the top boom in section through the rivets, showing the connection between the plates *b*, *b* and angle-iron *c*, *c*, *c*, *c*, and also between *b*, *b* and the angle-iron *e*...*e*. The figures are drawn to a scale of $\frac{1}{12}$.

The number and thickness, and therefore the total depth, of the plates forming the booms, as also the proportions of the other parts, must of course depend upon the *span* of the girder and the weight, live or dead, or both, it has to carry. The girder under discussion is 33 feet long, 2 feet 6 inches deep, and 1 foot 6 inches across the booms; the web-plates are $\frac{1}{4}$ inch thick, the top and bottom plates forming the booms are $\frac{1}{2}$ inch thick, the cover plates are $\frac{5}{8}$ inch thick, and the angle-iron *c* is $4 \times 4 \times \frac{1}{2}$ inches; the strut plates *f* and end plates *g* are $\frac{1}{4}$ inch thick, and the angle-iron *c* $3 \times 3 \times \frac{1}{2}$ inch.

183. As another example of a riveted structure we give that of a portable boiler, a portion of which is shown in

figs. 46 and 47, Plate XXIX. The boiler consists of an outer shell *a*, flue *c* inside the shell *d*, etc., but it is only with the shell and flue and their connection that we are concerned. The shell is built up of rings of plates, the end rings have attached to them end plates *b*, *b* bent to the shape shown in fig. 47. The shell and flue are cylindrical in form, but are not concentric, hence the end plates *b*, *b* are of the form shown in fig. 47, *e* being the centre for the shell and *f* that of the flue. The shell *a* and end plates *b*, *b* are connected by lap-joints, single riveted, as shown in fig. 46, which is a section through the central plane *e g* of fig. 47, of the front end of the boiler. The flue projects beyond the shell and end plate; round this projecting part is fixed a ring *d* of angle-iron, which is riveted to the flue and end plate, thus connecting the shell, flue, and end plate. Fig. 47 is an end elevation, with a portion of the angle-iron, ring, and flue in section, showing the rivets; and with a portion of the shell and end plate in section. The figures are drawn to a scale of $\frac{1}{12}$.

184. Gibs and Cotters.—There still remains another method of connecting two pieces to be described and illustrated—*gib* and *cotter*. This arrangement is employed both as a permanent and temporary connection; it is a connection for special purposes, and is not applicable to those already considered, except in the case of the cotter bolt. Sometimes one of the connected pieces is required to move while the other remains stationary, frequently both pieces have motion imparted to them, as in the case of the connecting rod of a steam engine, when the connection at the end is often made by means of gibs, a cotter, and a strap. Again, both connected pieces may be stationary, still in this case the principle of the connection is the same, as will be seen from the figures and descriptions immediately following.

There are three forms of this connection to describe:—(*a*) The simple cotter without gibs; (*b*) a cotter and one gib; (*c*) a cotter and two gibs. Of the second and third forms there are a variety of designs, and various means are employed to force home the cotter and keep it there.

A cotter or cutter is a tapered piece of metal, generally resembling in form and action a wedge, but with this difference, the wedge is used to force asunder parts of the same

piece or of different pieces, while the cotter is employed to draw together, by means of available parts, two or more pieces of metal. The amount of taper given to the cotter must not exceed the angle of repose of metal upon metal, which, for greased surfaces, may be taken at about 4°; a common rule is to make the taper $\frac{1}{2}$ to $\frac{3}{4}$ of an inch to each foot of length.

(*a.*) Cotter connecting two pieces without a gib. In the annexed figure we have an example of the use of a cotter without a gib being employed in conjunction with it; it here maintains the bolt *c* in the hole made in the piece *b* to receive it. The action of the cotter is simply to wedge itself tightly into the pieces, and maintain its hold by the grip thus induced. It is quite evident that so long as *d* keeps its place, the bolt *c* cannot be removed from *b*. In figs. 2 and 3, Plate XXIII., we have a very good example of a cotter connection. The cotter in this case is employed to connect the rod K H with the strap V of the eccentric, by forcing home the cotter, the collar on the rod H is forced against that on the strap V, clearance being allowed, as shown in fig. 3, to admit of this motion of the cotter, the action here is exactly similar to that shown in the former part of this paragraph, except that in the latter the collar H bears, or is forced against the other connected piece, while in the former the end *g* of the bolt *c* bears against the end of the hole in the piece *g*, which is at once perceived by an examination of the figure.

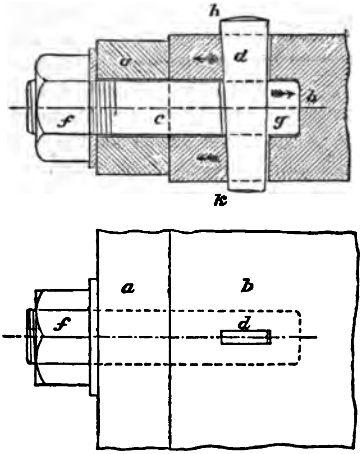


Fig. 178.

(*b.*) A cotter connecting two pieces when one gib is used. By referring to Plate XL., figs. 1 and 2, two views of a connecting rod are there given; a very fair example will there

be found of the *strap*, *gib*, and *cotter;* *o* is the cotter, *n n* the gib, and H the strap, the references refer to both figures. The shape of the gib *n n* is easily traced in the upper figure, the dotted line showing the part hidden from view. The object of making the gib in the form shown, is to render it impossible that it shall fall out when the connecting rod is at work. The cotter needs no further reference, it is seen how it assists to keep the gib in its place. The strap H is also shown completely; but, perhaps, a moment's inspection of the figure in the margin will give the student a better idea of its true shape. It must be understood that the strap here given is not the same shape as the one in fig. 2, Plate XL., to which our remarks above refer.

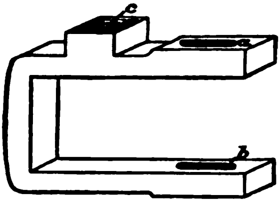


Fig. 179.

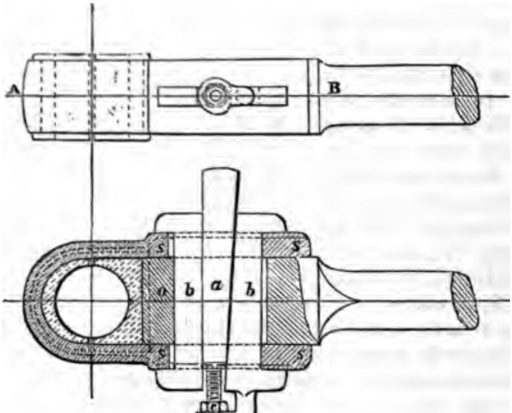


Fig. 180.

(*c.*) A cotter and two gibs. Fig. 180 shows a connecting head, held together by a strap, cotter, and two gibs; the

strap is marked *s s*, the cotter *a*, and the two gibs *b b*, respectively, they firmly hold the brasses at the end of the rod in their places; the advantage of two gibs is, that they keep the strap firmer against the brasses. The screw which forms the lower part of the gib is to prevent the cotter from falling or being jerked out when the engine is in motion. After the strap *ss* is put on the connecting rod, the gib *b* is inserted, and then the nut *c* is placed so that when the key *a* is put in, the nut can be screwed up. The key is driven home with the hammer, the nut *c* being slackened to allow it to come down. When it is made as tight as is required, the nut *d* is put on and screwed up tightly. Then *c* is screwed down, and so the two prevent the key from becoming slack. The hole for the bolt at *e* must be made elliptical, so as to allow the key to come down without bending the bolt.

SECTION III.

ROTATING PIECES AND THEIR BEARINGS—SHAFTING—KEYS—BEARINGS—BUSHES—PEDESTALS—PIVOTS—FOOTSTEPS—SLIDES OR MOTION BLOCKS—COUPLINGS—CLUTCHES.

185. In Chapter V., Section I., we have defined the kinds of simple moving pieces and their bearings; we now propose to give some examples of them in this and the following section.

186. Shafting.—The term shafting is applied to the lengths of material, usually of a circular cross section, which are employed for the purpose of transmitting motion by means of wheels, pulleys, or drums, etc., from one part of a mill or workshop to another. A *length*, or *line of shafting*, may consist of one piece, or of several lengths or pieces connected together by means of couplings. If the length or piece is short, say 5 or 6 feet and under, it is called a *shaft;* this is also the term applied to shafting employed in machines, where generally the shafts are short compared with a line of shafting in a mill or factory. The term *spindle* is often

employed instead of shaft; thus we say the spindle of a *lathe*, and not the shaft; and in many machines and machine tools the term spindle is used for a certain shaft, of which we shall have several examples when speaking of machine tools.

The cross section of all shafting is usually circular, as the form offers least resistance to the air; there are, however, other forms employed, as square, octagonal, etc. Shafts and shafting are generally solid, but in some cases hollow shafts are employed.

The stresses to which shafts and shafting are subjected are generally of a complicated kind, but there are certain stresses which most shafts have more or less to withstand, and to these we will refer. The material chiefly employed is wrought-iron; cast-iron and steel are also used. The relative strengths of these materials have been already given, and from what was then said the student can form an idea as to the best material to employ. The strain to which shafting is chiefly subjected tends to wrench the shaft asunder, or causes back-lash in the machinery driven, so that it runs unevenly. The remedies may be one of two—either make the shafting of greater diameter, or put a moderately-heavy fly-wheel on the end of the shaft; the wheel equalises the power. Some shafts are simply loaded transversely, and are only subject to a bending strain. Shafting that transmits power is subject merely to torsion, while crank shafts are subject to torsion and bending.

187. Collars and Necks of Shafts.—Shafts are supported in bearings, of which there are at least two for each; in the case of a line of shafting there are are several bearings, arranged at equal distances apart, or according to the special circumstances of the case. In all cases shafts have to be maintained in their places and prevented from leaving their bearings in the direction of their length; this is effected in several ways, as follows:—(1) By collars, *e c*, *welded* to the shaft, as shown in figs. 181 and 182. (2) By loose collars, as *d*, fig. 183, attached to the shafts by means of set-screws. (3) By the wheels, pulleys, etc., fixed on them, the *bosses* of which take the place of the fixed or loose collars before named. (4) [illegible] by a combination of these arrangements.

Figs. 181 and 182 represent an ordinary shaft with two fixed collars, *c* and *c*; the shaft is shown *broken off* at *a b*, which is the usual way of representing a piece of material, or a portion of a machine, which cannot be drawn in full according to scale; the length is always marked in figures, as shown above. The length of the neck *e*, which forms one of the bearings, is generally made $1\frac{1}{2}$ times the diameter of the shaft, for shafts under 6 inches diameter; some engineers

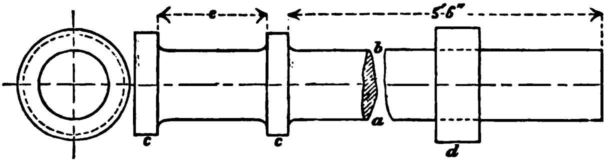


Fig. 181. Fig. 182. Fig. 183.

allow as much as two diameters, and in special cases, as, for example, in high-speed shafting, even more than this. One rule is, that for cast-iron shafting the right-hand figure journals must be $1\frac{1}{2}$ times the diameter of the shaft, and for wrought-iron $1\frac{3}{4}$ times the diameter. The following short table, taken from Sir William Fairbairn's works, gives an idea of the size of bearings and the weights they support:—

TABLE IX.

	Size.	Area of Bearing.	Weight on Bearing.	Weight per sq. in. on Bearing.
		sq. in.	lbs.	lbs.
Fly-wheel Shaft, wrought-iron,	18 × 14	252	45,024	178·21
Vertical Shaft, cast-iron,.........	11 × 8·6	95	25,061	242·7
Horizontal Shaft, cast-iron,.....	15 × 10	150	6,000	40·0
,, ,, wrought-iron,	6 × 3	18	540	30·0
,, ,, ,, ...	2 × 4	8	160	20·0

Loose collars are employed chiefly in cases where fixed collars would prevent the fixing of wheels, pulleys, etc.; *d*, fig. 185, represents such a collar. In figs. 184 and 185 the loose collar, *d*, is shown drawn to a larger scale; the figure

is partly in section, showing the set or pinching-screw *f*, by which it is fixed to the shaft.

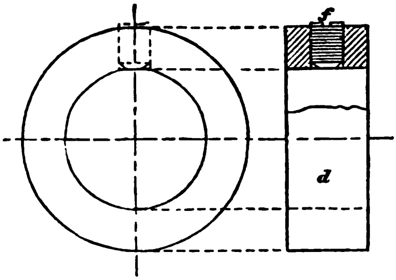


Fig. 184. Fig. 185.

188. Strength of Shafting.—The chief stress to which shafting is subjected is torsion, but in many cases we must consider the resistance it offers to deflection, and also the stiffness of the shafting. These elements vary in different cases; thus a short shaft, perfectly strong enough to transmit a certain amount of power, may be quite unequal to the task if it were, say, 5 times as long, the failure in this case resulting from a want of stiffness, or from not being strong enough to resist the lateral pressure due to one or more pulleys and their straps. It must be assumed that with shafting the deflection ought never to exceed $\frac{1}{100}$ part of an inch for every foot length of the shaft, for stiffness is a much more important element than strength.

We will first consider the torsional strength of shafting. All shafts and shafting must be made strong enough to resist torsional stress, and generally, in the case of short shafts, such as those used to carry the gearing of cranes, and shafts in machine tools, etc., this will be sufficient for the other stresses; that is to say, if the shafts in the cases named are strong enough to resist torsional stress, they will be strong enough to resist lateral stress, or are sufficiently stiff for all purposes. The amount of power a shaft is capable of transmitting also depends on the velocity at which it rotates, and also upon the kind of motion it transmits; if the shaft is the crank-shaft of a steam engine, then the *maximum* strain

thrown upon it must be ascertained, for it is clear the strain is not uniform throughout the stroke.

It is often necessary to have some part of a shaft larger in diameter than the rest, in which cases it is necessary to notice that the strength of the shaft is the strength of its smallest diameter. And the strength will not even equal this if the smaller and larger portions form at their junction a sharp angle, as any sudden enlargement tends to weaken the shaft; and in all cases where there is an enlargement, either for a collar, boss, or for other purposes, the change from the one to the other should be gradual, and formed by a curved outline, as shown in fig. 182. This rounding of the corners in journals increases their strength one-fifth.

189. The elements to be considered with respect to torsion are—the strength of the material, the direction of the shaft, and the twisting force; these we have already spoken of in Art. 140, to which the student's attention is again directed. The simplest form in which we can describe twisting force is as one due to a weight acting at the end of a lever, and tending to twist the shaft, as shown in that paragraph. Of course, the strength of different metals to resist torsion varies very much. Formerly, timber was chiefly employed to transmit power subject to torsion only; this was succeeded by cast-iron, as in mill work, but both are now generally superseded by wrought-iron.

Deflection or *bending* is the stress due to a force acting transversely to the shaft, such as arises from the weight of the gearing, pulleys, etc., together with the weight of the shaft itself, which exercises an influence not always to be neglected. The relative strength of a shaft, girder, beam, etc., so far as it depends upon its supports and the manner in which the load is distributed, either uniformly or collected at one point, is expressed very simply.

	Relative Strength.
When supported at one end and loaded at the other, ...	=1
,, ,, and load distributed, ...	=2
,, both ends and loaded at the centre, ...	=4
,, ,, and load equally distributed,	=8

The strength is increased considerably (according to some authorities the numbers above become from 12 to 16) by *firmly*

fixing the shaft at both ends, when it is entirely under different conditions. In short shafts the deflection can be entirely disregarded, but in the case of shafting for mill work the deflecting stress must be taken into consideration, and the thickness so proportioned that the deflection shall not exceed the $\frac{1}{100}$ part of an inch per foot of length.

190. Keys for Shafts.—It is necessary to connect the wheels, pulleys or riggers, etc., with the shafts that carry them; this is done by means of *keys*, which are pieces of metal, usually steel, of a square or rectangular cross section. There are several methods of keying wheels to their shafts, and these depend chiefly on the nature of the work that has to be performed by the wheels, pulleys, etc. Thus, if the power to be transmitted is comparatively small in amount, then a system of keying may be employed which, if the opposite were the case, would result in the failure of the system.

We may divide the methods of keying into two classes:— I. Where only one key is employed for connecting each wheel or pulley to its shaft. II. Where four or more keys are thus employed, as in wheels that are *staked* to their shafts.

The same division holds good if we say, for Class I., the wheels, pulleys, etc., are *bored* and the shafts *turned;* and for II. that, as a rule, the wheels, etc., are not bored, or the shafts turned.

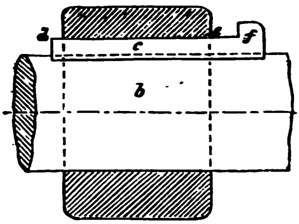


Fig. 187.

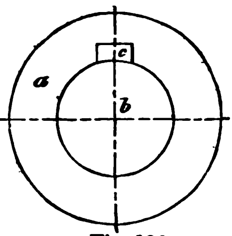


Fig. 186.

191. Hollow or Saddle Keys.—In the first class, which is by far the most common, we have the *hollow key*, which fits the shaft without the latter being cut in any way; such a key is shown in figs. 186 and 187.

The surface of the key next to the shaft is hollowed out a little, so as to fit the curved surface of the shaft, as shown in fig. 186, where *a* is the boss of the wheel or pulley, *b* the shaft, and *c* the key, which in this example we have shown with a head, *f*, to facilitate its being *drawn* easily.

Fig. 187 shows the boss in section; the figures are drawn to a scale of $\frac{1}{4}$. This form of key can only be employed in cases where the power to be transmitted is comparatively small, as the efficiency of the key depends upon the friction between the surface of the key and that of the shaft in contact with it. It is necessary to give a little taper to the key in the direction *d e*, so that the full amount of friction may be made available. And it will be clear that, if we are to have a tight fit between the shaft and key, the latter must fit easily on the two edges or sides that are in contact with the *key-way* or *key-bed* of the wheel or pulley.

192. Keys for Shafts with Flats.—Figs. 188 and 189 represent a very common form of key arrangement, which may be employed for all classes of work except the heaviest, and such special work as will be considered later on. This is a much firmer arrangement than the one described in the previous article, but it necessitates the removal of a portion of the shaft.

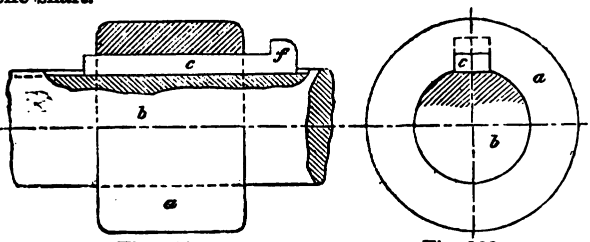


Fig. 189. Fig. 188.

If the wheel or pulley has a fixed position, then it is only necessary to make a *flat* or plain key-bed on that portion of the shaft where the wheel is to be keyed; the flat should be longer than the key, and should gradually merge into the full diameter of the shaft, as shown in fig. 189. The same letters of reference are employed in this example as in the previous one. As in the preceding case, the key must be

slightly tapered in the direction $d\,e$, and must fit easily on the sides of the key-way of the wheel or pulley.

193. Another method of keying, where plain surfaces are employed, is shown in fig. 190; in this case four keys are employed for round or square shafts, and in some cases, for very heavy work, where square shafts are used, eight keys may be employed. This mode of fixing or keying wheels to their shafts is termed *staking*, and a wheel so fixed is said to be *staked*. Staking is generally only employed in cases where the shaft is not turned, and the hole or eye of the wheel or pulley is not bored. Four flat surfaces are prepared on the shaft, and a similar number in the eye of the wheel, to receive the keys, which are of a rectangular cross section, the dimensions of which may be taken from Table X., p. 165. The upper portion of the figure represents a round shaft, and the lower a square one. If the shaft is square, and eight keys are employed, they should be fixed near to the angles of the square, one on each side. The bosses are marked *a*, *a*, the shafts *b*, *b*, the keys *c*, *c*, and the space between the inner surface of the boss and the shaft *d*, *d*. This was the method formerly employed to secure paddle-wheel shaft centres in their proper place.

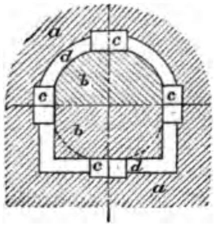


Fig. 190.

194. Cone Keys.—Cone keys are employed when a wheel has to be bored sufficiently large to pass over a boss on the shaft. They are of cast-iron with parting plates, nearly dividing each into three pieces. The casting is bored and turned and broken into three pieces, which are chipped to clear away the metal where the fractures take place; thus each cone becomes three tapering keys of the dimensions required to fix the pulley or wheel firmly on the shaft.

195. Sunk Keys.—In figs. 191 and 192 is shown the usual method of keying where the power to be transmitted is considerable; it is superior to either of the previous examples; although a more costly arrangement, it is at the same time a more perfect one, but it tends to weaken the shaft. The shaft has a groove cut in it, in the direction of its length,

to receive the key, and the wheel or pulley has a similar groove cut in it, as shown in the figures. If the position of the wheel or pulley is fixed, then the groove in the shaft need only be long enough to admit of connecting and disconnecting the wheel by the key; if the wheel is on the end of a shaft, then the groove may be a little longer than the key.

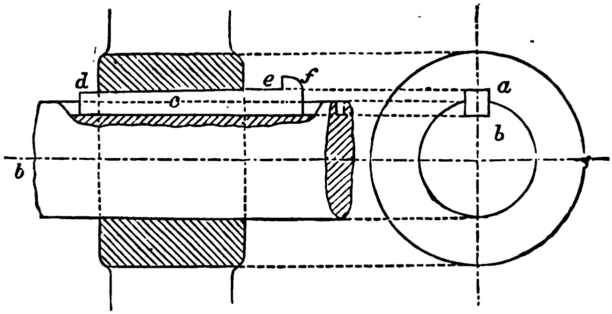


Fig. 191. Fig. 192.

Fig. 191 is a section of the boss *a* of a wheel or pulley, showing the key *c* in position after being driven home; the shaft *b* is also partly in section, showing the groove in it; the key has a head *f*, by means of which and a suitable lever or *key-drawer* it may be easily withdrawn: heads on keys are very convenient in fitting them in their places.

196. We may note here, that there are two different opinions among engineers as to the most efficient way of *fixing* the key in the present arrangement. Some maintain that it is best to have the key to fit tightly on the *sides*, that is on the vertical surfaces in fig. 192 of the key-way, and to fit easily on the top face, where the key and the broad surface of the boss are in contact; while others maintain that the key ought to fight tightly on the two broad surfaces, one in the shaft and the other in the wheel, and to fit easily on the sides of the key-way. We do not intend to enter into this question at length, as space forbids it, but simply to give as our opinion, that the latter mode of fixing the key is the better arrangement; that is a key fitting easily

on the sides of the key-way, and tightly on the top and bottom; the key having a little taper in the direction $d\,e$, fig. 191, to admit of its being driven home tight. The amount of taper should not exceed $\frac{1}{8}$ inch per foot in length. By driving a key home tight we simply mean, supposing it to have a very slight taper, that just as much force is to be used as will permanently fix it in position and no more; the amount of force required will, of course, depend upon the size of the pieces to be connected.

197. Sliding Keys.—It is sometimes necessary for either a wheel to slide upon its shaft, or for the shaft to slide through the wheel, which remains stationary; in such cases it is clear the key cannot be fixed in any of the ways previously described.

To meet such cases as these there are employed *sliding keys*, of which there are two common forms. The key is attached to the wheel or pulley, and fits easily the key-way in the shaft, so as to admit of the motions before named.

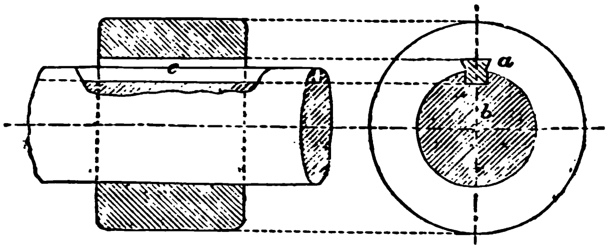


Fig. 193. Fig. 194.

In figs. 193 and 194 we have shown one mode of fixing a sliding key; in this case the key c is *dovetailed* into the boss a of the wheel; the cross section of the key is shown in fig. 194. The key is driven in tight into the boss of the wheel to prevent it leaving its position when fixed, and sometimes a screw is employed to maintain it in that position.

198. In figs. 195, 196, and 197 is shown another form of sliding key; fig. 195 is a sectional elevation, fig. 196 an end elevation, and fig. 197 a plan of the key.

The key in this case has a cylindrical head of a diameter equal to the breadth of the key, as shown in fig. 197, which

fits into a hole of an equal diameter in the boss, as shown in fig. 195. An ordinary key-way is formed in the boss to receive the key, so that the head of the key has simply to maintain the whole in position; while the strain is thrown upon the sides of the key-way, as it is previously explained in the example shown in figs. 191 and 192.

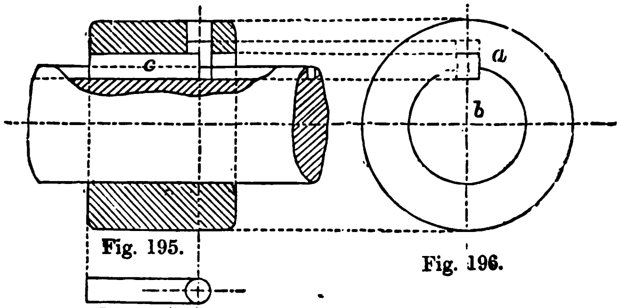


Fig. 195.

Fig. 196.

Fig. 197.

In the two examples just shown, *a* is the boss of the wheel or pulley, *b* the shaft, and *c* the key. The figures are drawn to a scale of $\frac{1}{4}$.

199. Dimensions of Keys.—As the form of keying illustrated in figs. 191 and 192 is the most common, we give in Table X. the dimensions of such keys for shafts from $\frac{1}{2}$ inch to 12 inches diameter; the dimensions are for the two common forms of keys in use, viz., square and rectangular. The amount of taper given to keys in the direction of their length differs a little, but about $\frac{1}{8}$ inch per foot is usual; allowance should be made in the bottom of the key-way of the *wheel* or *pulley* for this taper, that is to say, the key-way must be *slotted* or *chipped* with this amount of taper in the direction of its length; the bottom of the key-way in the shaft being parallel with a plane containing the axis.

There is a little difference among engineers as to the dimensions of keys, those we have given are pretty generally employed. The same dimensions of rectangular keys given in Table X. may be used for keys employed in staking wheels, also for most of the other kinds.

The keys are sunk half in the shaft and half in the boss

of the wheel, the distance being measured along the line A B, fig. 198; that is to say, the distances *e* and *f* are equal.

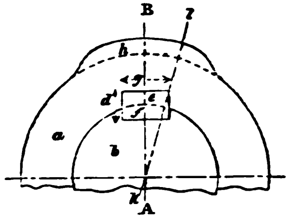


Fig. 198.

It is necessary to strengthen the boss where the key-way is to be cut, as shown at *h*, so that the boss shall be equally strong in every part. As a rule, no section of the boss taken through any part of the key-way, as the section *kl*, should have a section less in area than any other part of the boss where there is no key-way; this section should, in fact, have a greater area than any other, on account of the angles at the bottom of the key-way.

The principle of the key is that of a wedge, and it secures the wheel in position mainly by that power, observation alone, as already hinted, determines the taper or draught. Steel is preferable for small keys, but in some situations soft iron is best, for it hugs the shaft closer than the harder metal. The taper determines the hold of the key, and very little force is necessary to drive it; if we exceed a certain limit, then the power of the wedge is exerted to split the wheel and cause it to work off.

200. Robertson's Wedge Keys.—In fig. 199 is shown a method of keying introduced by Mr. Robertson, the inventor of the frictional gearing, which we shall hereafter endeavour to explain; it is an epitome of frictional gearing in form and principle.

Fig. 199 represents a shaft *b*, $2\frac{1}{4}$ inches diameter with its key *c*. The shaft has three grooves cut in it, to receive the three wedge-shaped projections on the lower surface of the key. By this arrangement there is less metal cut out of the shaft than in the ordinary sunk key arrangement, which is an advantage, and the strength of the shaft is not diminished.

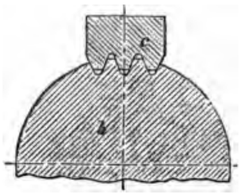


Fig. 199.

[illegible]tor claims for this method of keying increased

hold between the key and shaft, and that it obviates the necessity of forming enlarged parts, swellings, or collars, on shafts, for the purpose of fixing heavy wheels, couplings, etc. This, of course, being due to the fact that less metal is cut away from the shaft, and also on account of the increased amount of surfaces in contact, and the form of those surfaces.

TABLE X.

DIMENSIONS OF KEYS FOR SHAFTS.

All Dimensions are in Inches and Decimals of an Inch.

Diameter of Shaft.	Size of Key, Section, a Square.	Sizes of Key, Section, a Rectangle.	
		Breadth (g).	Depth (d).
·5	·2	·25	·2
·75	·25	·312	·227
1·0	·31	·375	·25
1·25	·34	·437	·27
1·50	·375	·5	·295
1·75	·406	·562	·317
2·0	·437	·625	·34
2·25	·468	·687	·36
2·50	·5	·75	·385
2·75	·562	·812	·40
3·0	·577	·875	·43
3·5	·64	1·0	·475
4·0	·687	1·125	·52
4·5	·75	1·25	·565
5·0	·81	1·375	·61
6·0	1·0	1·625	·71
to 12·0	Add ·0625 in. for every ·5 in. diameter of shaft.	Add ·125 in. for every ·5 in. diameter of shaft.	Add ·045 in. for every ·5 in. diameter of shaft.

BEARINGS AND THEIR SUPPORTS.

We have previously defined the term bearing, and have also stated the kinds of bearings in general use, and their forms; we now propose to give a few examples of these bearings and their supporting pieces, considering, as in the definition, the bearings to be the surfaces of contact. In this section we shall consider the bearings of sliding and of

rotating pieces, and, in the next section, pieces having helical motion.

The supporting pieces for horizontal rotating pieces are *journals*, *bushes*, *pedestals* or *plummer-blocks*, etc.; for vertical rotating pieces we have, in addition, *footsteps* for supporting the pivots or ends of the rotating pieces.

The supporting pieces for horizontal or vertical sliding pieces are *slides*, *slide-bars*, etc.

201. The extent of the bearing surfaces must depend upon the amount of pressure exerted upon those surfaces; when it exceeds a certain amount per unit of area the friction increases rapidly, owing to the ungent being squeezed out, which causes the two surfaces to come into contact, and, as a consequence, the bearings become heated and abrasion of the surfaces ensues. According to experiments made by Rennie, friction increases rapidly when the pressure per square inch exceeds 616 lbs., in the case of wrought-iron or steel on cast-iron, and for cast-iron on brass when it exceeds 784 lbs. But these are limits which should be and are avoided in practice.

202. Journal Bearings.—The term journal is frequently applied to bearings that are formed either in the frame of the machine, or are directly carried by it, although in some cases the term is applied to the bearings of pedestals, etc. The pieces that form the bearing surfaces are called *steps* or *brasses*, and are made of steel, cast-iron, brass, or other alloy, according to the circumstances of the case. Properly, we should speak of *journals* and *journal bearings*.

Generally, each bearing consists of two steps, a top one and a bottom one, the surfaces of contact of the two being either in the plane that contains the axis of the rotating piece, or in separate planes at a short distance from and parallel with this plane. In the latter case the space between the steps is left for the purpose of *tightening-up* as the metal wears. As similar steps are used for pedestals, we shall refer to them more fully under the head of pedestals.

203. Bushes or Sleeves.—A bush is a hollow cylinder of metal, steel, cast-iron, brass, or other alloy, in which a shaft or other rotating piece rotates, the inner surface of the bush forming the bearing. Bushes are generally fixed in the

frame of the machine; they differ from the steps or brasses described in the previous paragraph in one respect; although they are used for similar purposes they are in one piece, whereas the steps are in two. The object of a bush is to supply two requirements—first, a suitable material, which may be and generally is quite different to that of the frame, for the bearing of the rotating piece; and, second, a bearing which may easily be replaced by another when worn, which could not be the case if the frame itself formed the bearing.

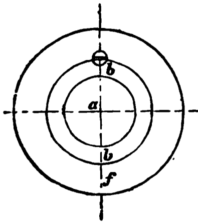


Fig. 200.

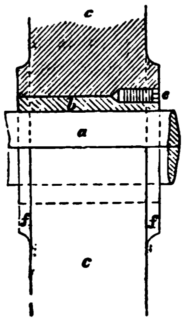


Fig. 201.

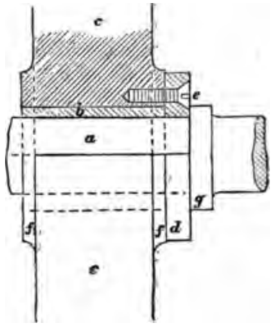


Fig. 202.

A common form of bush is shown in figs. 200 and 201; it consists simply of a hollow cylinder *b*, turned on its outer surfaces, and fitting accurately a bored hole in the frame *c*, to which it is attached by the screw *e*; the object of this screw is to prevent the bush leaving the hole, or turning round in it. The bush is bored to receive the rotating piece *a*, and is turned on its ends so as to be flush with its facings, *ff*, of the frame.

204. Another form of bush or sleeve is shown in fig. 202; the only difference between it and the preceding one is, that it has a collar *d* on one end, which may be used for fixing it, by means of a screw or screws *e*, to the frame. This is a better form of bush for some purposes, as when the shaft has a collar, *g*, on it, in which case the collar of the bush forms a suitable end or side bearing for it; or if, instead of a collar, there is the boss of a wheel or pulley bearing against the bush;

in these cases the collar of the bush, which can be easily replaced, is worn and not the machine.

We have stated that a bush is a hollow *cylinder*, but it may have any form suitable for the purpose; as it is a bearing for rotating pieces, its surfaces, inside and outside, must be surfaces of revolution, as those of cylinders, cones, etc. The inner surface need not necessarily be of the same form as the outer; for instance, one may be cylindrical and the other conical. Where the wear is considerable it is not advisable to use bushes, unless they can be turned round a little as they wear, or be replaced readily, as they soon get out of truth. The common plan is to use movable steps, which admit of adjustment to compensate for wear. Many examples may be observed in the Plates accompanying this work.

205. Slides or Motion Blocks and Guides.—A very common form of parallel motion introduced of late years, for horizontal engines, is found in the slide or motion block and its guides. The principle is, that the head of the piston-rod is made to slide in one or two grooves formed in or fixed on the bed-plate. A favourite way is to make the guide part of the bed-plate, to secure the greatest amount of steadiness.

Figs. 203, 204, and 205 illustrate the form of these blocks. Fig. 203 is a front sectional elevation, fig. 204 an end elevation, and fig. 205 a plan—all drawn to a scale of 2 inches to the foot, the letters of each figure corresponding to those in the other. Our example is taken from a large horizontal engine. At *b* are seen screws, by means of which the block can be adjusted; at *a a'* are two round nuts, which are fixed by screwing down the check-nuts *d* and *d'*; pieces, as seen in the figure at *a* and *a'*, are cut out of these nuts to take a suitable spanner. By means of these the adjustment is easily effected, and the centre of the block kept at exactly the same height as the centre of the cylinder. The shoes, A and D B, are loose, and only kept on by the guide *c*; *i.e.*, if the whole were lifted out of the guides the shoes would fall off. The brasses are adjusted by the stud E, which is screwed into the block, and then *h*, a check-nut, is tightened down in the same way as *d* and *d'*. The connecting-rod is fastened in

such a manner to a pin as to make the pin move with it. After the pin is put in, a hole is drilled and made to take the tapered pin *e e*; the hole passes half through the pin and half through the rod-eye, as seen at *e e*. Consider for a moment that if the leverage is too great, or if the centre line E F be too high or too low, that as the piston moves from left to right, there will be a tendency to throw the friction on the points A or B, while in the return stroke it will come on D or G, hence the necessity for the most careful adjustment; but when so made there is absolutely no jerking on the brasses, the whole working with the most astonishing smoothness.

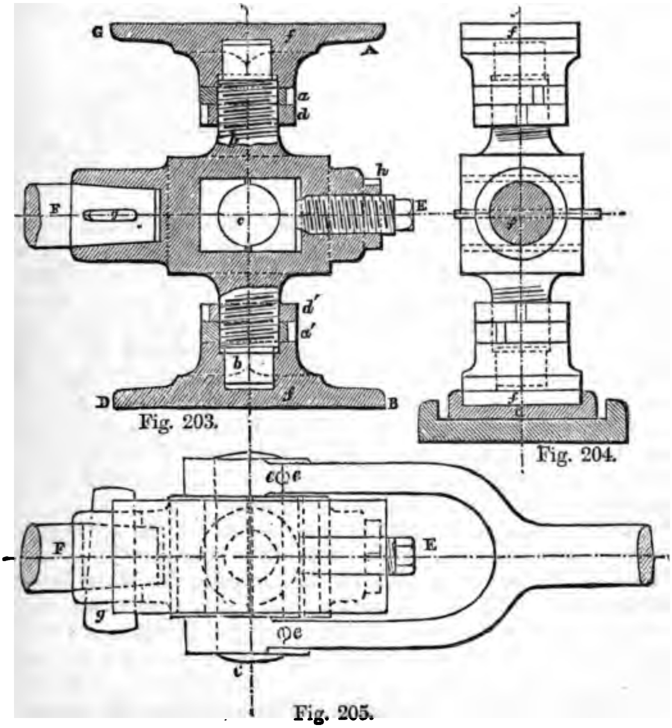


Fig. 203.

Fig. 204.

Fig. 205.

One form of guides, within which a slide block works, is shown below, but it is not the guide suited for the above block. Fig. 206 is a front elevation; the head of the connecting-rod is seen at A, its motion being from *a* to *b*. Fig. 207 is a plan of either part B or C, the dotted lines showing the groove in which, in this case, the block A moves.

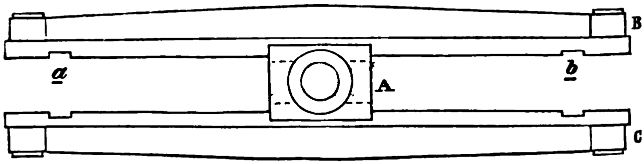


Fig. 206.

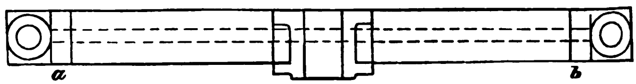


Fig. 207.

206. Pedestal or Plummer-block.—The supporting piece of a bearing, which does not form part of the frame of a machine, and can be detached from it, is called a *pedestal*, *pillow-block*, or *plummer-block*. In figs. 1, 2, and 3, Plate XVIII., we illustrate one form of pedestal, and in figs. 1 to 13, Plates XIX. and XX., we show this pedestal in detail. Figs. 1, 2, and 3, Plate XVIII., are examples of finished shade-line drawings; fig. 2 is a front elevation, fig. 1 a plan, and fig. 3 an end elevation. We shall have occasion to refer to these figures later on. Figs. 1, 2, 3, 6, and 7, Plate XIX., are respectively plan, front elevations, half of which is in section, and sectional and end elevations.

207. Pivots and Footsteps.—Vertical shafts having to carry heavy gearing, or to transmit a considerable power, have to be provided with an end bearing for their lower extremities. The end of such a shaft forms a pivot which is supported by a footstep. In figs. 4, 5, and 6, Plate XVIII., we show one form of footstep; fig. 4 is a plan, fig. 5 a sectional, and fig. 6 an end elevation. The footstep consists of a cast-iron frame C attached to the base plate D, the connec-

tion between the two being strengthened by the feathers E E. The portion C is square in plan, with a square hole in it to receive a square block F of cast-iron, which carries the bearing G, which is made of a material suitable for the purpose. The base plate is provided with bolt holes, through which bolts *b b* pass for the purpose of fixing it to its foundation by the nuts *c c*. The block F can be raised or lowered according to circumstances, to allow for wear of the bearing G, or for its renewal by means of a wedge H, one end *d* of which passes through a snug *f*, and is screwed to receive the nut *e*, by means of which it is held in position, and drawn to and fro as circumstances require. The lower end of the wedge is parallel with the surface of the base plate, upon which there is a chipping piece *g* for the edge of the wedge to bear on, while its upper edge is tapered and is in contact with the under side of the block; this block is provided with chipping pieces *h h* at its angles, to reduce the amount of labour in fitting. The pivot K is carried by the bearing piece G, which is bored out to receive its turned end, this end is slightly curved. There are various forms of footsteps, but in all cases the object is the same, to provide a suitable support and adjustable bearing for the ends of vertical shafts.

208. Footstep Bearing.—In machinery it is frequently necessary to employ a footstep bearing; vertical shafting requires a footstep for its lower end to rest in, to prevent lateral motion, and to keep the vertical adjustment correct. The footsteps for turbines present a technical difficulty, as provision must be made for lubrication under water, which is done by providing an oil casing connected with proper pipes. Sometimes a lignum vitæ bearing is employed, which is an excellent method; for when metal works on wood, water is a very good lubricant.

We have here, in fig. 209, an example of a readily adjustable footstep bearing; A is a cast-iron stand turned so as to receive the brass B, within which the shaft S works, resting on the piece of steel D at the bottom of the recess. The whole rests on the cast-iron plate C, which may be of any shape to suit taste or convenience. The piece of steel D is made very hard; as it wears it is raised by means of *a*, which is screwed into the bottom of the cast-iron stand D. A hole is left in the bottom

piece C large enough to receive the bolt head, and to allow it to move up and have a box key on it at the same time. The brass is prevented from turning round in its seat either by being fitted into an hexagonal or octagonal recess, or else by using the pin *b*, which is tightly fitted into a round hole drilled in the brass, while its projecting end fits into a recess in the cast-iron cut to receive it. This pin is called a "key," "feather," "snug," etc., in different localities. Lateral adjustment is provided for by arranging for the bottom piece to shift as required, or else by keeping D A in its place by set screws, when the whole arrangement has to be modified.

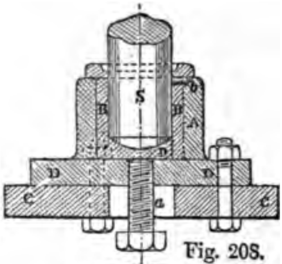


Fig. 208.

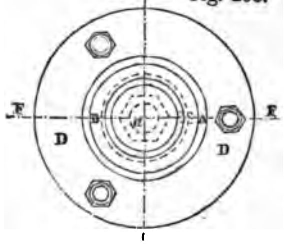


Fig. 209.

209. Stuffing-Boxes and Glands.—The covers of steam engine cylinders, and those of other vessels containing a fluid or gas, are generally provided with stuffing-boxes and glands. In figs. 210 and 211 we have shown a common form of stuffing-box and gland, C is the cylinder cover which forms a bearing for the spindle or rod D; cast on to the cover is the stuffing-box E with its flange F. The stuffing-box is bored to receive the cylindrical portion G of the gland, on the outside of which is cast a flange H through which passes two or more bolts K; the gland is bored to receive the rod D, to which it acts as a bearing. Between the inner end of the gland G and the bottom of the stuffing-box E, at P, is placed the packing which is to render the joint water, gas, steam, or air tight. By means of the bolts K K and the nuts L L the position of the gland is regulated, which is necessary to compensate for the wear of the packing and also for its renewal; A is a bush or lining. Stuffing-boxes

or glands are made in a variety of forms, but their object is always the same, and that is to provide a suitable bearing and gas-tight joint for spindles or rods working through them, as the piston-rods of steam engines, the rods of slide and other valves. The upper figure is half in elevation and half in section, the whole being drawn to a scale of two inches to one foot. Other examples will be found in Plates XXXIX., XL., XLVI., XLVII., etc.

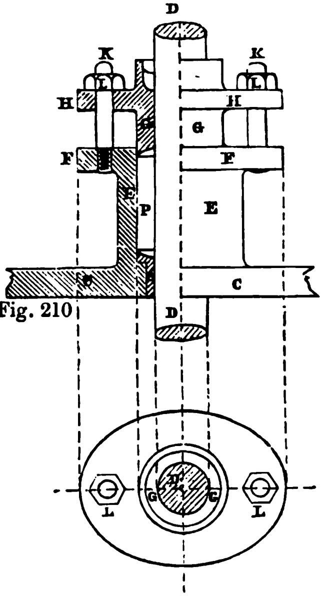


Fig. 210

Fig. 211.

COUPLINGS—CLUTCHES.

210. When long lengths of shafting are employed, it is necessary to provide the means of connecting the individual pieces that form a line or length. Also, in some instances, it is necessary to have the power and means to connect and disconnect short lengths of shafting. These connections may be either permanent or temporary, according to the circumstances of the case; if they are permanent they are generally called couplings, and if otherwise, clutches. Of the former class we have the box, the face-plate, and the friction coupling; of the latter, there are clutches with from two teeth upwards, friction cones, wedge and groove, friction clutches, etc.

211. Butt Box Coupling.—In figs. 212 to 214 is shown an ordinary butt box coupling; fig. 214 is a plan, fig. 213 an elevation, with the box and portion of the shaft in section, and fig. 212 is a front elevation. The ends *a* and *b* of the shafts are swelled out or increased in diameter to receive the box *c*; this is done for two purposes: to enable the box to

pass over any fixed collars that may be on the shafts, and also to maintain the strength of the shafts, which would not be the case if there were no enlargements, and the key-ways for the connecting key *d* were cut in the smaller diameter of the shaft. This key-way connects the three pieces *a*, *b*, and *c*, the ends *a* and *b* are turned, and the box *c* bored, the key fitting in key-ways made in them. On the shaft *a* is shown a bearing *e*; it is usual to fix couplings near to a bearing.

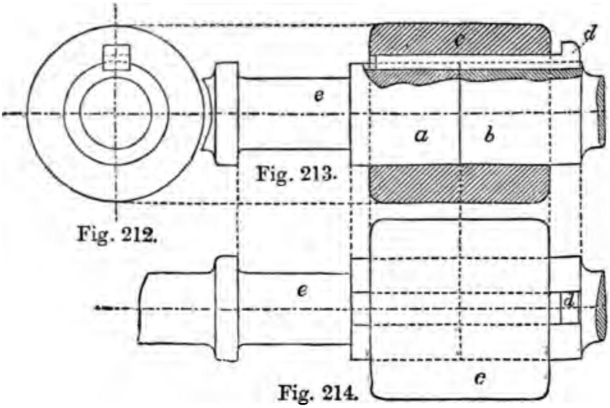


Fig. 212. Fig. 213. Fig. 214.

212. Half-lap Box Coupling.—A good form of coupling for shafts up to about six inches diameter is shown in figs. 215, 216, and 217; fig. 217 is a plan, fig. 216 an elevation, with the box and portion of the shafts in section, and fig. 215 an end elevation. This coupling was introduced by Mr. Fairbairn, and has proved a very efficient one. It will be observed that the chief difference between this one and the last is in the lap *a b*, which is made equal to the diameter of the shaft. The following are the proportions given by the inventor:—

Area of coupling - - - -	= 2 × area of the shaft.
Or, in other words, diameter of coupling - - - -	= 1·4142 × diameter of shaft.
Length of lap - - - -	= diameter of shaft.
Length of box - - - -	= 2 × diameter of shaft.
To which may be added outside diameter of box - -	= $2\frac{1}{4}$ × diameter of shaft.

There is more workmanship required in fitting together this coupling than the last, but it is far superior to it; in the previous one, the efficiency of the arrangement depends on the key, while in the present case the key is only a secondary piece, and is used simply to keep the ends *a* and *b* of the shafts and the box *c* united, the chief strain is borne by the surfaces in contact, as seen in fig. 216. The figure represents a coupling adapted for a shaft three inches in diameter, drawn to a scale of $\frac{1}{8}$.

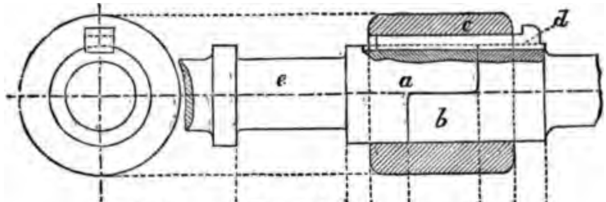


Fig. 215. Fig. 216.

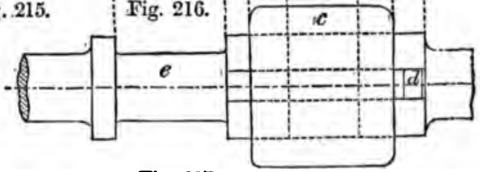
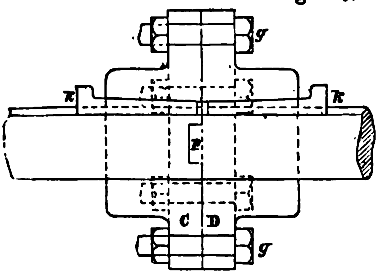


Fig. 217.



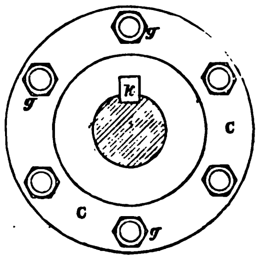


Fig. 218. Fig. 219.

213. Disc or Flange Coupling.—In figs. 218 and 219 is shown a form of coupling used for both large and small shafts, but more generally for shafts six inches and under in diameter.

The coupling consists of two face-plates or flanges C and D, with bosses, and one being keyed to the end of each shaft. These face-plates are connected by bolts g. The end of one shaft at p enters a short distance into the face-plate, keyed by k k or k on the other shaft so as to maintain the axes of the two shafts in line. These shafts are turned, and the coupling bored to receive them, and generally the whole of each half coupling is turned after being fixed in position. The objection to this form of coupling is that the strain has to be borne by the keys; the two face-plates being connected by bolts, there is no danger in that direction, as we can easily make the bolt connection of sufficient strength; hence the keying should be carefully done. The number and diameter of the bolts vary with different makers and under different circumstances, the total area of the bolts should vary between ·33 and ·5 of the area of either shaft. In some cases flanges are forged on the ends of the shafts to be connected, and the two flanges connected by bolts, as in the case just mentioned. This method is more expensive than the other, but at the same time a more perfect one, as the connecting keys are dispensed with and the size of the flanges much reduced in proportion, as they are made of the same material as the shaft itself.

214. Wedge and Groove Coupling.—Another form of disc coupling is the wedge and groove coupling. In this case the inner faces of the flanges have grooves cut in them, so that they present a series of concentric grooves and wedges, the wedges of the one fitting the grooves in the other. The form of the grooves resembles that employed by Mr. Robertson in his friction gearing, and the coupling under notice was introduced by him. Each half of the coupling is keyed to its shaft, and then the two halves connected by bolts, as in the previous case.

215. Claw Coupling and Clutch.—The connecting piece we are about to describe may be employed either as a permanent coupling or as a clutch. The figures represent it as a clutch. Fig. 222 is a plan; fig. 221 a sectional elevation showing the claws A and B in section, and also portions of the shafts; fig. 220 is an end elevation. When employed as a coupling there are two equal halves similar to B, one on each of the

shafts *c* and *d*, each half being fixed to its shaft by a key. The inner face of each half is provided with three claws, and three recesses to receive the claws of the other half, thus forming an efficient connection; but of course the strain has first to be received and borne by the keys, and hence the same objection exists as in previous cases. This form of coupling is well adapted for long shafts, as the amount of metal necessary to produce a firm connection is less than in the one described in the article on flange coupling. When used as a coupling, that is, as a permanent connection, each half is fixed to its shaft by a key or keys.

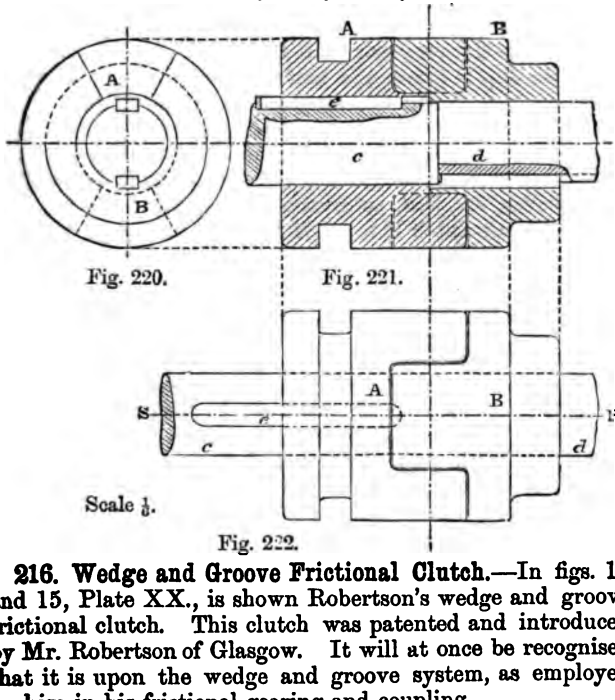


Fig. 220. Fig. 221.

Scale ⅛.

Fig. 222.

216. Wedge and Groove Frictional Clutch.—In figs. 14 and 15, Plate XX., is shown Robertson's wedge and groove frictional clutch. This clutch was patented and introduced by Mr. Robertson of Glasgow. It will at once be recognised that it is upon the wedge and groove system, as employed by him in his frictional gearing and coupling.

Fig. 15 is a plan, partly in section, showing the clutch; fig.

14 is an elevation, also partly in section, the figures show the means of engaging and disengaging the clutch. On the shaft B, and free to turn with or without it, is one-half the clutch marked D; upon the boss of this half clutch is fixed the wheel C, which it is required to engage and disengage with the shaft B. The other half E of the clutch is attached to the shaft B by two keys *b b*, but it is free to slide upon the shaft. The boss of this half-coupling is provided with collars F F, which fits in grooves in a clip G; this clip is provided with a circular pin H, which fits in an eccentric boss K. The boss K is carried in a bearing N, which is supported by a foot M resting on the top face of N, and attached to the boss K is a lever L. By means of the lever L the eccentric boss K can be turned; and as it turns, the clip G moves the half-coupling in or out of gear; in the position shown the coupling is "in gear," but if the lever were moved into the position indicated by *l* it would then be "out of gear." The two half-clutches D and E have their inner surfaces provided with concentric circular wedges and grooves, *c*, *c*, *d*, *d*, so arranged that, when the lever L forces the half-clutch E towards D, so as to engage with it, the wedges of the one enter the corresponding grooves of the other, and prodnce a firm wedging contact. The wedges and grooves are kept slightly lubricated, so as to prevent all abrasion of their surfaces. There are many modifications of this clutch in use, but the one we have described is a common form. The diameter of the clutch is made seven times the diameter of the necks of the shaft; that is, a wrought-iron shaft 2 inches diameter would require a clutch 14 inches diameter. With this proportion the clutch and shaft are equal as regards resistance to torsion; the clutch would be able to resist the same amount of torsion as the shaft. Such an arrangement is excessively efficient, and no slipping whatever takes place.

217. Cone Clutch.—The cone clutch was invented so that the workman might have the power of disconnecting a line of shafting expeditiously; in fact all clutches are intended to answer this purpose more readily than couplings. A B C D is a solid piece of metal in conical shape, which admits of a lateral motion from left to the right on the shaft P; E F G H

is a corresponding piece, into which A B C D will slide; as soon as the one slides within the other the force of friction between the surfaces in contact is sufficient to engage the line of shafting. E F G H is firmly keyed on to the shaft O; O and P are brought close together at *a*. A lever acting at A D impels the one part of the clutch against the other.

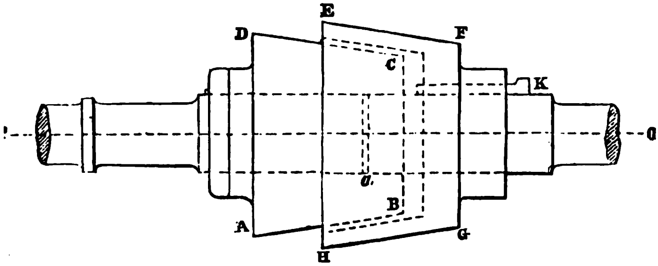


Fig. 223.

A double-cone clutch has been invented by Mr. Sellers. It consists of an outside "muff," which is placed over the ends of the shafts. In the barrel are placed two conical bushes, which are jammed into their place by three screw bolts. The grasp of these bushes upon the shaft act as a first-rate coupling.

Difference between a Clutch and Coupling.—Couplings are divided into two classes—(*a*) *Couplings* (proper); (*b*) *Clutches.* The chief difference between the two is, that coupling is a permanent connection, or a clutch can be disconnected at once, but a coupling requires much time. Couplings are permanent clutches.

SECTION IV.

HELICAL CURVES—SCREWS—NUTS.

218. Helical or Screw Curves.—Having in a former paragraph defined helical or screw motion, we now proceed first to show how to draw the helical curves, and then the helical surfaces as represented by screws.

Figs. 1 and 2, Plate VII.—The helical curve is traced as follows:—If, during the revolution of a cylinder, a marker, which moves parallel to the axis of the cylinder and at a uniform rate, traces upon its surface a curve, the curve so traced is called the *helical* or *screw curve*. The distance moved through by the marker, during one revolution of the cylinder, is termed the *pitch*, and the direction in which it moves determines whether the curve is *right* or *left handed*. Assuming the cylinder to be turning in the direction of the hands of a watch, as indicated in fig. 1, which we have previously defined as right-handed rotation, and the marker to move from right to left (0—16, fig. 2), the curve is *right-handed*, and *left-handed* if *vice versâ*.

The curves shown in figs. 1 and 2 are right-handed and of the same pitch, but differing in diameter; the pitch is the distance *a b* (0...16). In the example given the curves are either supposed to be fine *wire* bent to the required form, or else the cylinders upon which they are traced are supposed to be transparent, so that the back half of the curve may be seen. The front half of the curve is marked 0...8, the back half 8...16. If the curve were a left-handed one, the portion marked 8...16 would be the front, and that marked 0...8 the back half.

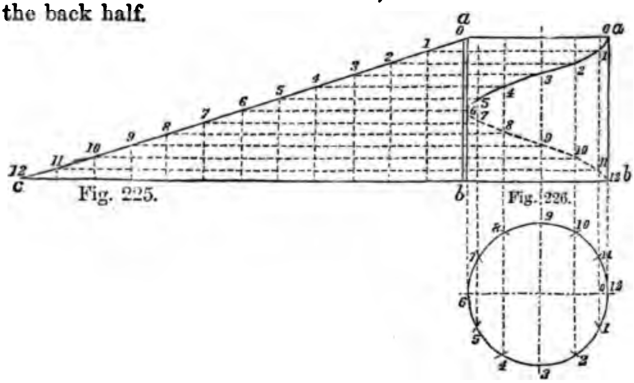


Fig. 225.

Fig. 226.

Fig. 224.

The length of the curve is equal to the hypothenuse of a right-angled triangle, *a b c*, fig. 225, having for its base the

circumference of the cylinder, and for its height the pitch. Figs. 224 and 226 represent a cylinder upon which is traced one turn of the helix; $a\,b$ is the pitch; $a\,c$, fig. 225, is a *development* of the curve; and the angle $a\,c\,b$ is the angle of the curve; in screws the angle $a\,c\,b$ is the *angle of the thread.*

The drawing of the curve is as follows, the pitch and the diameter of the cylinder being given :—We will take the larger curve first. Having drawn the centre lines and the projections of the cylinder, figs. 1 and 2, Plate VII., divide the circumference of the circle, fig. 1, into any convenient number of equal parts, divisible by 4, as 12 or 16, and the pitch $a\,b$, fig. 2, into the same number of equal parts. In the example we have used 16. Number these points 1, 2, etc., to 16, respectively, in both figures. From 0, 1, 2, etc., to 8, fig. 1, draw lines parallel to the axis C C; and from 1, 2, etc., to 16, fig. 2, draw lines perpendicular to the axis. Then the intersections of the lines 1...1, 2...2, 3...3, are points in the curve; join these points and the curve is complete.

In the top half of fig. 2, between b and d, is shown a quarter of the larger curve, numbered 4 to 8. In the bottom half, between b and d, is shown a quarter of the smaller curve, numbered 0 to 4, which can be obtained similarly, the construction lines indicating clearly how to project it. In these examples we have taken enough points to determine the curve with a sufficient degree of accuracy for ordinary purposes; if greater accuracy is required it can be obtained by taking a larger number of points on the circumference of the circle, and the same number between a and b. It will be noticed the curve is *quickest* between 0...2 and 6...8; intermediate points may be taken between these points to determine the curve more accurately.

219. Screws.—Screws are made by cutting helical grooves, of a triangular, square, or other cross section,* in cylindrical pieces of metal or wood; the ridge or projecting part is termed the *thread*, and the hollow the *space;* the *pitch* is the distance between two consecutive threads, measured as previously described.

The two most common kinds of screws in use, excepting

* A section made by a plane perpendicular to the direction of the length of the groove.

those for wood, are the V or triangular, and square-headed; the former chiefly used for bolts, studs, and set-screws, the latter to transmit pressure, as in *screw-presses*, and to transmit motion to the *slides* of *lathes*, *planing machines*, and other engineering tools.

The V and the square-threaded screws are shown in Plates VII. and VIII., the drawing of which we shall refer to presently.

The form of the thread depends upon the kind of work for which the screw is to be employed; and from the examples just given it will be seen there is a considerable distinction in the use to which each form is applied. The square thread is the better form to use for purposes where the wear is considerable; that is to say, for transmitting motion and force, each transmission being repeated often. The V thread is a better form to use for screws for connecting or fastening purposes, as it has an advantage over the square thread in strength when employed for such purposes.

220. Right and Left Handed Screws.—Screws are right or left handed according to the direction in which the nut moves when the screw is turned round in a right-handed direction, or as defined, perhaps, more clearly above. Screws are considered to be right-handed single thread unless otherwise stated. Left-handed screws are only used in special cases.

221. V-threaded Screws.—A V-threaded screw, $2\frac{1}{4}$ inches diameter and $\frac{1}{4}$ inch pitch, is represented by figs. 3, 4, and 13, 14, Plate VII. It is usual to denote the pitch, which varies according to the diameter of the screw, by so many threads per inch in length; in the example shown the screw has four threads per inch, equivalent to $\frac{1}{4}$ inch pitch. The form of V thread now in general use is the "*Whitworth Screw Thread.*"* In fig. 227 is shown a longitudinal section of this thread. The distance $a\,a$ is the pitch; the angles $a'a\,a'$, $a\,a'a$ are each 55°, so that the depth $a\,e = \cdot 887$, $a\,a' = \cdot 96\ aa$,† that is, assuming the thread to be angular, but $\frac{1}{6}$ of

* Introduced by Sir Joseph Whitworth, Bart.

† $a\,e = a\,a' \times \text{cosine } 27\frac{1}{2}^\circ = a\,a' \times \cdot 887$;
also $a'e = \frac{1}{2}\,a'b' = \frac{1}{2}\,a\,a$, and $\therefore a\,e = a'e \times \text{tangent } 62\frac{1}{2}^\circ = \frac{1}{2}aa \times \text{tan. } 62\frac{1}{2}^\circ = \frac{1}{2}\,a\,a \times 1\cdot 92 = \cdot 96\ a\,a$.
If $a\,a = 1$ inch, $a\,e = \cdot 96$ inch.

the depth is rounded off at the top and at the bottom, as shown, leaving only $\frac{4}{6}$ of $a\,e$ ($= \cdot 64\ a\,a$) as the depth of the rounded head.

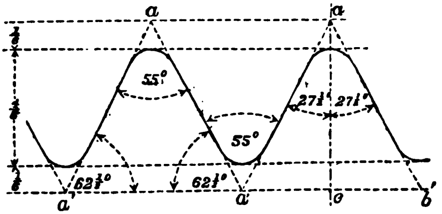


Fig. 227.

The following table contains a list of the number of threads per inch in length for screws from $\frac{1}{16}$ inch to 6 inches diameter, according to the *Whitworth Standard*:—

TABLE XI.

Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.
$\frac{1}{16}$	60	$\frac{5}{8}$	11	$1\frac{3}{4}$	5	$3\frac{1}{2}$	$3\frac{1}{4}$
$\frac{3}{32}$	48	$\frac{11}{16}$	11	$1\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{3}{4}$	3
$\frac{1}{8}$	40	$\frac{3}{4}$	10	2	$4\frac{1}{2}$	4	3
$\frac{5}{32}$	32	$\frac{13}{16}$	10	$2\frac{1}{8}$	$4\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{7}{8}$
$\frac{3}{16}$	24	$\frac{7}{8}$	9	$2\frac{1}{4}$	4	$4\frac{1}{2}$	$2\frac{7}{8}$
$\frac{7}{32}$	24	$\frac{15}{16}$	9	$2\frac{3}{8}$	4	$4\frac{3}{4}$	$2\frac{3}{4}$
$\frac{1}{4}$	20	1	8	$2\frac{1}{2}$	4	5	$2\frac{3}{4}$
$\frac{5}{16}$	18	$1\frac{1}{8}$	7	$2\frac{5}{8}$	4	$5\frac{1}{4}$	$2\frac{5}{8}$
$\frac{3}{8}$	16	$1\frac{1}{4}$	7	$2\frac{3}{4}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$2\frac{5}{8}$
$\frac{7}{16}$	14	$1\frac{3}{8}$	6	$2\frac{7}{8}$	$3\frac{1}{2}$	$5\frac{3}{4}$	$2\frac{1}{2}$
$\frac{1}{2}$	12	$1\frac{1}{2}$	6	3	$3\frac{1}{2}$	6	$2\frac{1}{2}$
$\frac{9}{16}$	12	$1\frac{5}{8}$	5	$3\frac{1}{4}$	$3\frac{1}{4}$		

222. Figs. 3 and 4, Plate VII.—The drawing of the screw is as follows:—Draw the centre lines, and the outline of the cylinder of $2\frac{1}{4}$ inches diameter, upon which the screw is to be cut; and set off the pitch $a\,b$ along the centre line C C, or upon the outline of the cylinder, as shown at 4...4,

$a \ldots b$, fig. 4. Having thus divided the screw for the pitch, draw $a\,a'$, $a\,b'$, so that $a'a\,b'$ contains an angle of 55°, $a\,a'$, $a\,b'$ being equally inclined to the axis; from b draw $b\,b'$ parallel to $a\,a'$ meeting $a\,b'$ in b', then b' is the bottom of the groove. Draw $b'\,4'$ parallel to the axis, meeting the centre line of fig. 3 in $4'$; with C $4'$ as a radius, describe the semicircle $4'2'0'$, which will represent the bottom of the groove or thread. The groove $a\,b'b$ is termed the *space*, and is occupied by a projecting thread in the nut. The curves 4 b, 4 a, a a, etc., which form the tops of the threads, and $a'a'$, $b'b'$, etc., which form the bottoms of the threads, are obtained in the same manner as described for the helix.

In this example we have divided the semicircle, which forms half the end elevation, into four equal parts, and, therefore, the pitch into eight equal parts. As each curve in making a revolution passes through a space $a\,b$, half the curve, as seen in the figure, will have passed through the space $c\,b = \frac{1}{2}\,a\,b$; the latter curve is numbered $0 \ldots 4$. In drawing the V we may either draw $a\,a'$ inclined to the axis at $62\frac{1}{2}°$ $(90° - \frac{55°}{2})$ by setting off the angle by means of a protractor from a horizontal line, as the axis, or by placing the protractor at a, perpendicular to the axis, and marking off a line $a\,a'$ inclined to $a\,e$, fig. 227, at $27\frac{1}{2}°$ $(\frac{55°}{2})$, $a\,b'$ being drawn in a similar manner.

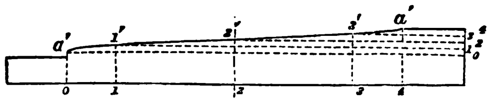


Fig. 228.

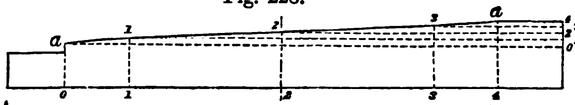


Fig. 229.

Having determined the curves for the top and bottom of the thread, as shown by the dotted lines on the left hand of fig. 4, the remaining curves may be drawn by means of *templates*, consisting of thin wood or cardboard cut to the required form. The templates for the curves 4—b, 4—a, etc., are shown in figs. 228 and 229. It is much better to make

separate templates for the different curves we require, than to try and make use of the ordinary *moulds* or *curves;* this remark applies to all cases where there is a repetition of the same curve in a drawing.

In the example, we have shown the thread of the screw with angular top and bottom; this, however, is not quite correct, as we have already stated, but for convenience in drawing we may assume it to be so.

223. V-threaded Screw and Nut.—As previously stated, the bearings of screws are *nuts* which fit the former accurately. Figs. 13 and 14, Plate VII., are respectively end and longitudinal sectional elevation of a nut for a V-threaded screw of $2\frac{1}{4}$ inches diameter, and four threads per inch; in the right-hand half of fig. 14 is shown the screw, a portion of which is in section showing the screw and nut in contact. It will be noticed that the curves for the top and bottom of the thread, in the portion of the nut shown, are inclined in the opposite direction to those of the screw, and that is because the figure shows the threads of the nut for the back half of the screw, while the threads of the screw shown are for the front half. The dotted lines show the tops and bottoms of the thread on the back of the screw, and these are parallel to those of the nut, as should be the case.

The drawing of these figures calls for no special remark, as the principles employed are exactly similar to those of the previous example, to which we refer the student.

224. Square-threaded Screws.—Figs. 15 and 16, Plate VIII., represent in plan and elevation a right-handed square-threaded screw, $2\frac{1}{2}$ inches diameter, and having two threads per inch, or $\frac{1}{2}$ inch pitch. A section of the thread of the screw made by a plane passing through S P, is a square whose side $= \frac{1}{2}$ the pitch; the space being also a square of equal side. The thread and space, therefore, make up the pitch; but this refers only to *single-threaded* screws, hence it is necessary to define the pitch more accurately, which we will endeavour to do. We shall define the term *pitch of a screw,* so that it is independent of the number of threads on the screw, which we consider to be the clearest manner of expressing it. In all cases either the screw or the nut is fixed, and prevented from moving lengthwise (in the direction

of the axis of the screw); we shall consider the nut to be the moving piece, as being most suitable for the definition.

225. The Pitch of a Screw is the distance moved through by the nut during one revolution of the screw. Of course, if we consider the nut to be fixed, it will be seen at once that the pitch is the distance, measured in a direction parallel to the axis, moved through by the screw during one revolution. We may have one or more threads on a screw; therefore, to find the size or thickness of the thread for square-threaded screws, divide the pitch by twice the number of threads on the screw, and the quotient will be the required size. Thus, in fig. 16, $a\,b =$ the pitch $= \frac{1}{2}$ inch, therefore the thickness of the thread $= \frac{1}{2}\,a\,b = \frac{1}{4}$ inch; in fig. 21 there are two threads on the screw, which is 1 inch pitch, therefore the thickness of the thread $= \frac{1}{4}$ inch.

A right-handed double square-threaded screw, $2\frac{1}{2}$ inches diameter, 1 inch pitch, is shown in figs. 19, 20, and 21, Plate VIII. The side of the square, which is a cross section of the thread, is $\frac{1}{4}$ of the pitch $= \frac{1}{4}$ inch, as just stated.

226. The distinction between right and left handed screws applies also to square-threaded screws. For square-threaded screws there is no strict standard for the number of threads per inch of length, according to the diameter of the screw, as there is for V-threaded screws. In some establishments the rule is, for the same diameter of screw, to allow the number of threads per inch to be one-half that of the V-threaded screw. This rule agrees very nearly with the following table:—

TABLE XII.

Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.	Dia. of Screw.	No. of Threads per in.
$\frac{1}{4}$	10	$\frac{5}{8}$	7	1	5	$1\frac{3}{4}$	$2\frac{1}{2}$
$\frac{5}{16}$	10	$\frac{11}{16}$	7	$1\frac{1}{8}$	4	$1\frac{7}{8}$	$2\frac{1}{4}$
$\frac{3}{8}$	9	$\frac{3}{4}$	6	$1\frac{1}{4}$	$3\frac{1}{2}$	2	$2\frac{1}{4}$
$\frac{7}{16}$	8	$\frac{13}{16}$	6	$1\frac{3}{8}$	3	$2\frac{1}{8}$	$2\frac{1}{4}$
$\frac{1}{2}$	7	$\frac{7}{8}$	6	$1\frac{1}{2}$	3	$2\frac{1}{4}$	2
$\frac{9}{16}$	7	$\frac{15}{16}$	6	$1\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{1}{2}$	2

227. We now refer to the drawing of the screws shown in figs. 15 and 16, Plate VIII. Draw the centre lines, and the projections of the cylinder of $2\frac{1}{2}$ inches diameter, upon which the screw is to be cut; set off the pitch *a b*, and obtain a curve of half a revolution for the top of the thread, as 0...6. Take half the pitch and set it off either along the centre line C C, or upon the outline of the cylinder, commencing from the curve already drawn; and draw from each of these points curves parallel to 0...6, as shown. Join 12...6, 0...0, *e*...*e*, etc., forming the tops of the thread, and draw the curves for the bottoms of the thread as shown by $0'1'2'3'$; these curves commence on the smaller cylinder in the same horizontal plane as those on the larger cylinder, as shown on 6, 6, fig. 16. The construction lines show how to complete the figure. A portion of the back half of the thread is shown by the dotted lines *e f*, *e f*, portions of which, *f g*, *e h*, are in full where they cross the space. It will be noticed that these lines are inclined in the opposite direction to the lines representing the front half of the thread. The following figures show the construction of the curves more clearly. The curves for these screws may be drawn by means of templates, as previously explained.

Figs. 20 and 21, Plate VIII.—These figures represent a right-handed double square-threaded screw, $2\frac{1}{2}$ inches diameter, 1 inch pitch. Having drawn the centre lines and the outline of the cylinder, set off the pitch *a b* and divide it into four equal parts, as there are two separate threads on the screw, and draw the curves for the tops and bottoms of the threads, as shown by the construction lines. As similar letters of reference are employed for these figures, we need only refer to the previous figures for any information required.

228. Nut for Square-threaded Screw.—We have already defined a nut as the bearing of a screw, we now show in figs. 17 and 18, Plate VIII., the nut for the screw represented in figs. 15 and 16. Fig. 17 is a half plan, and fig. 18 a sectional elevation. The drawing of these figures requires no explanation, as the lines are exactly similar to those of the screw, but of course the curved lines are inclined in the opposite direction to the full lines in the screw. Fig. 19 is a longitudinal section of the threads of the screw and nut,

showing them in contact. Brass and cast-iron are the materials chiefly employed for nuts when they are used as bearings; but in the case of nuts for bolts, wrought-iron is generally employed.

229. Approximate Methods of Drawing Screw Threads. —We have hitherto shown how to draw the true form of the threads of screws, excepting the case of V-threaded screws; however, in most instances, approximations to the true form are employed, and, generally, the smaller the scale of the drawing, the further from the true form the approximations are carried. We shall now illustrate some of these approximations.

Figs. 5 and 6, Plate VII., represent the V-threaded screw shown in figs. 3 and 4, drawn to a scale of $\frac{1}{2}$; the curved lines *a a*, *a′a′* are here replaced by straight lines; this is the only approximation made, and it will be seen that it approaches the true form very nearly, as shown in the full size figures. Figs. 7 and 8 represent figs. 3 and 4 drawn to a scale of $\frac{1}{4}$; in this case the Vs are not shown, and the approximation is, of course, not so good as the previous one. In still smaller scale-drawings, lines are used to represent the tops of the thread only.

Figs. 9 and 10 represent the right-handed double square-threaded screw shown in figs. 20 and 21, Plate VIII., drawn to a scale of $\frac{1}{2}$; the curved lines being replaced by straight ones. This is a good approximation, as it contains all the lines in the correct figure, the only substitution being straight lines for curved ones. Figs. 11 and 12 represent a left-handed square-threaded screw, $2\frac{1}{2}$ inches diameter, $\frac{1}{2}$ inch pitch, drawn to a scale of $\frac{1}{4}$. The approximation here shown is a very common one, and for small scale-drawings may be used with advantage; if the scale is too small to show even this much, then straight lines representing the tops of the thread may be used, as in the case of V-threaded screws.

CHAPTER VII.

ON THE TRANSMISSION OF MOTION BY WHEEL-WORK.

SECTION I.

WHEELS IN GENERAL—SPUR WHEELS—PITCH—RACK AND PINION.

230. THE kind or form of wheels to be employed in transmitting motion from one rotating piece to another, will depend upon the kind of motion to be transmitted and the relative position of the axes; generally, we have a choice of arrangements, but often this choice is restricted by other considerations, which we need not examine at present. The wheels employed for transmitting motion are usually provided with *teeth* to ensure regularity of motion, and to enable them to transmit a greater force than could be done conveniently with toothless or friction wheels. We shall assume in the first instance that the wheels are toothless, and that we have therefore smooth surfaces in contact, and that when one wheel turns it transmits its motion to the one in contact with it, the two surfaces *rolling* together without *sliding*. The surfaces which thus roll together become the *pitch surfaces* in toothed wheels, and the object to be kept in view when designing the teeth is that this rolling motion of the pitch surfaces shall be maintained as nearly as possible.

231. The constructions and calculations, which we shall employ in the following articles, apply equally to wheels with teeth and without; for in all cases they will be referred to the pitch surfaces, or, as we shall term them, *pitch lines*, or *pitch circles* in the case of circular wheels. The pitch line is the trace of the pitch surface upon a plane at right angles to that surface; that is, in the case of circular and non-circular wheels, at right angles to the axis of rotation, and there-

fore parallel to the plane of rotation, as the face of the wheel; in the case of racks, at right angles to the face of the rack.

When wheels, or a wheel and rack, are so connected that if one turns the other turns also, the wheels are said to be *in gear*, and *out of gear* if *vice versâ*.

The term gear or gearing is usually applied to a combination of toothed wheels; it is also employed to denote a number of connected moving pieces other than wheels. We shall now give examples of the various kinds of wheels coming under the conditions here enumerated.

232. Spur Wheels.—Spur wheels are employed for transmitting rotary motion from one shaft to another, with a constant directional relation and a constant velocity-ratio; the shafts being parallel. There are two kinds of spur wheels, external and internal, according as the teeth are outside or inside the *rim*. In the first case, where we have two external wheels in gear, the wheels rotate in *opposite* directions; and in the second, where we have an external and an internal wheel in gear, they rotate in the *same* direction.

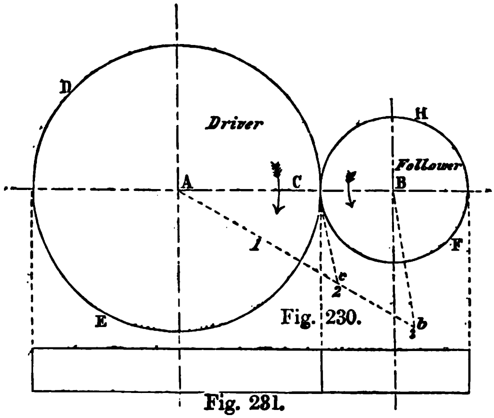


Fig. 230.

Fig. 231.

I. *External Wheels.*—A and B, fig. 230, are the centres of two shafts which are to be connected, so that B shall make two revolutions to one of A; required the diameters of the two wheels. Join A B and divide it in C, so that A C : B C

: : 2 : 1, then A C is the radius of the driver A, and B C that of the follower B. The line A B is called the *line of centres*, and the point of contact C of the pitch circles is always in that line.

The point C may be found by dividing A B into three equal parts by trial, as the division is a simple one; but as this is an exceptional case, we give a general method. Join A B and draw any line A b, making an angle of about 30° with A B, and upon it set off A c and $c\,b$, so that A c : $c\,b$: : the number of revolutions of A : the number of revolutions of B (in the example as 2 : 1). Join B b, and from c draw c C parallel to b B, cutting A B in C, then A C and B C are the required radii. The velocity-ratio of the axes A and B $= \frac{\text{C B}}{\text{A C}} = \frac{1}{2}$. Fig. 231 is a plan of the wheels.

II. *Internal and External Wheels.*—When it is necessary that the two connected shafts shall rotate in the same direction, an internal and external wheel are employed, the internal wheel is generally a *pinion;* that is, a wheel having a small number of teeth, usually from 10 to 20; the term pinion is employed in a relative sense, if the diameters of the two wheels differ considerably—say one is three or four times that of the other—then the smaller is ordinarily called a pinion.

A and B, fig. 232, are the centres of two shafts which are to be connected by spur wheels and are to rotate in the same direction; B is to make three revolutions to one of A. Join A B and produce it to C, making B C : A C : : 1 : 3, then A C and B C are the radii of the required wheels. The construction for finding C is similar to that in the previous example, except that C is in A B produced. In this example the velocity-ratio $= \frac{\text{C B}}{\text{A C}} = \frac{1}{3}$. In the first case, the distance AB between

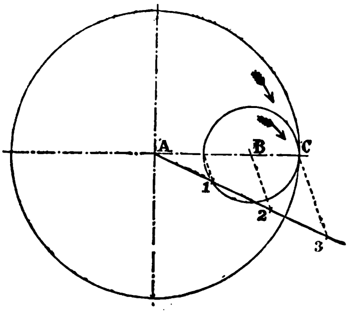


Fig. 232.

the centres of the wheels is equal to the *sum*, and in the second case to the *difference* of the radii of the wheels.

233. In the two examples just given, and in all cases of circular wheels, there are certain relations which regulate the proportions of the connecting pieces as regards radius, angular velocity, number of revolutions, and (number of) teeth.

In previous articles we stated the ordinary methods of expressing the rate of motion of rotating pieces, and in the following paragraphs we give examples of those methods. Let N and n denote the number of teeth, and R and r the radius of the driver and follower respectively; then

$$\frac{N}{n} = \frac{R}{r}.$$

Again, let V and v respectively denote the angular velocities, and E and e the number of revolutions of each in a given time; then

$$\frac{N}{n} = \frac{R}{r} = \frac{v}{V} = \frac{e}{E}.$$

Besides the above, we have to consider the pitch of the teeth of the wheels, and as most of our calculations involve this quantity, we shall therefore define the term and give some examples.

234. Pitch.—There are two methods of defining the pitch of a tooth; by the first, which is the most common, we speak of *circular pitch*, and by the second, which is used for small pitches only, of *diametral pitch*.

235. The Circular Pitch of a tooth is the distance, *measured along the pitch line*, from the centre of one tooth to the centre of the next; if the pitch line is a straight line, as in the case of a straight *rack*, the pitch is the distance, measured as before, along that straight line. The pitch is sometimes defined as the distance, measured as before, from the inside or outside of a tooth to the corresponding inside or outside of the next tooth, of course the length is the same in both cases. In fig. 234, page 198, the distance A D or f C represents the pitch, and in fig. 2, Plate XI., it is represented by g C or C n, etc. If the pitch line is a curved line, as a circle, the pitch is measured along the *arc* of the pitch line; that is, suppose the pitch is 1 inch, we mean the *arc*, and not the *chord* of the pitch line between the centres of two adjacent *teeth*, is 1 inch long. In the case of complete wheels it will

be readily understood that the pitch must be contained a whole number of times in the pitch line.

236. The following equation connects the pitch, the number of teeth, and the diameter of the pitch circle; let P denote the pitch, D the diameter of the pitch circle, and N the number of teeth of a wheel, P and D being given in inches and parts of an inch :—

$$\text{Then } P \times N = D \times \pi \text{ ......(1);}$$

which may be put in the forms—

$$N = \frac{\pi}{P}.D \text{......(2)}, \quad D = \frac{P}{\pi}.N \text{......(3)}, \quad \frac{D}{N} = \frac{P}{\pi} \text{......(4)}.$$

Equation (3) is the one most frequently required. The values of N and D vary for every different wheel, and are therefore very numerous, π is a constant ($= 3{\cdot}1416$), and P has few values, and they are definite compared with either N or D. The definite values of P usually employed are $\frac{1}{4}$ to 2. advancing by $\frac{1}{8}$, 2 to $3\frac{1}{2}$ advancing by $\frac{1}{4}$, etc. An examination of the following table will quickly manifest to the student the order in which these values advance. The values $\frac{\pi}{P}$ and $\frac{P}{\pi}$ in equations (2) and (3) are usually calculated for the above values of the pitch, and registered in a table of references so as to facilitate calculations; the table below is an example of such an one.

TABLE XIII.

(1). Pitch in inches.	(2). Values of $\frac{\pi}{P}$	(3). Values of $\frac{P}{\pi}$	(1). Pitch in inches.	(2). Values of $\frac{\pi}{P}$	(3). Values of $\frac{P}{\pi}$
5	0·6283	1·5915	$1\frac{5}{8}$	1·9264	0·5141
$4\frac{1}{2}$	0·6981	1·4270	$1\frac{1}{2}$	2·0944	0·4774
4	0·7854	1·2732	$1\frac{3}{8}$	2·2848	0·4377
$3\frac{1}{2}$	0·8976	1·1141	$1\frac{1}{4}$	2·5132	0·3978
$2\frac{1}{4}$	0·9666	1·0345	$1\frac{1}{8}$	2·7924	0·3580
3	1·0472	0·9548	1	3·1416	0·3182
$2\frac{3}{4}$	1·1333	0·8754	$\frac{7}{8}$	3·5904	0·2785
$2\frac{1}{2}$	1·2566	0·7958	$\frac{3}{4}$	4·1888	0·2386
$2\frac{1}{4}$	1·3963	0·7135	$\frac{5}{8}$	5·0265	0·1988
2	1·5708	0·6366	$\frac{1}{2}$	6·2832	0·1591
$1\frac{7}{8}$	1·6755	0·5937	$\frac{3}{8}$	8·3776	0·1194
$1\frac{3}{4}$	1·7952	0·5570	$\frac{1}{4}$	12·5664	0·0796

237. A few examples will illustrate the use of this table.

I. Required the diameter of the pitch circle of a wheel of 52 teeth, $1\frac{1}{4}$ inch pitch.

$$D = \frac{P}{\pi} \times N \ldots\ldots\ldots\ldots\ldots(3)$$

In column (3) the value of $\frac{P}{\pi}$ for $1\frac{1}{4}$ inch pitch is ·3978;

$$\therefore D = \cdot 3978 \times 52 = 20\cdot 6856 \text{ inches.}$$

II. Required the number of teeth in a wheel 11·14 inches diameter, $\frac{7}{8}$ inch pitch.

$$N = \frac{\pi}{P} \times D. \ldots\ldots\ldots\ldots\ldots(2)$$

In column (2) the value of $\frac{\pi}{P}$ for $\frac{7}{8}$ inch pitch is 3·5904;

$$\therefore N = 3\cdot 5904 \times 11\cdot 14 = 39\cdot 69 = 40 \text{ nearly.}$$

III. Required the pitch of a wheel of 65 teeth, 31 inches diameter.

$$\frac{D}{N} = \frac{P}{\pi} \ldots\ldots\ldots\ldots\ldots\ldots(4)$$

$$\frac{31}{65} = \cdot 477 \text{ nearly.}$$

This value of $\frac{P}{\pi}$ in the table under consideration corresponds nearly to $1\frac{1}{2}$ inch pitch.

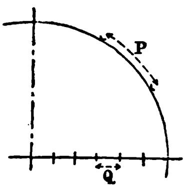


Fig. 233.

238. The Diametral Pitch for the teeth of a circular wheel is the length of a fraction of the *diameter* of the pitch circle, while the *circular pitch* is the length of a fraction of the *circumference;* thus, suppose the wheel in fig. 233 has twelve teeth, the circular pitch $= \frac{1}{12}$ of the circumference = P, the diametral pitch $= \frac{1}{12}$ of the diameter = Q.

$$\therefore Q : P :: 1 : 3\cdot 1416.$$

To find the diametral pitch divide the diameter of the pitch circle by the number of teeth; thus, let P, D, and N be the circular pitch, the diameter of the pitch circle, and

the number of teeth respectively, and Q the diametral pitch, then

$$Q=\frac{D}{N}\ldots\ldots\ldots\ldots(1) \qquad D=NQ\ldots\ldots\ldots\ldots(2)$$

$$\text{Also } \frac{D}{N}=\frac{P}{\pi}, \qquad \therefore\ Q=\frac{P}{\pi}\ldots\ldots\ldots\ldots(3).$$

Hence it is already seen by equation (3) that the diametral pitch is always exhibited in the last table under the heading $\frac{P}{\pi}$, but we presently give a table exhibiting it more readily.

239. Instead of using the diametral pitch in the form just given, it is usual to say a wheel has a certain number of teeth per inch of diameter; thus a wheel of four teeth per inch of diameter is called a *four-pitch* wheel; a wheel of nine teeth per inch a *nine-pitch* wheel, and so on.

The number of teeth per inch usually employed are 3, 4, 5, 6, 7, 8, 9, 10, 12, 14, 16, and 20.

Let q denote the number of teeth per inch of diameter, which is always a whole number, then $Q=\frac{1}{q}$, and equation (1) may be written

$$\frac{1}{q}=\frac{D}{N}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

Equation (3) becomes

$$\frac{1}{q}=\frac{P}{\pi}, \therefore\ P=\frac{\pi}{q}\ldots\ldots\ldots\ldots(5).$$

The values of P corresponding to the values of q are registered in Table XIV. for reference.

TABLE XIV.

Values of q.	P in decimals of an inch.	P to nearest $\frac{1}{32}$ of an in.	Values of q.	P in decimals of an inch.	P to nearest $\frac{1}{32}$ of an in.
3	1·047	1	9	·349	$\frac{11}{32}$
4	·785	$\frac{25}{32}$	10	·314	$\frac{9}{32}$
5	·628	$\frac{5}{8}$	12	·262	$\frac{1}{4}$
6	·524	$\frac{17}{32}$	14	·224	$\frac{7}{32}$
7	·449	$\frac{7}{16}$	16	·196	$\frac{3}{16}$
8	·393	$\frac{13}{32}$	20	·157	$\frac{5}{32}$

240. The following examples will explain the use of this table:—

I. What is the diameter of a five-pitch wheel of twenty teeth, and what is its circular pitch?

$$\frac{D}{N}=\frac{1}{q},\quad D=\frac{N}{q}=\frac{20}{5}=4 \text{ inches.}$$

$$P=\frac{\pi}{q}=\cdot 628=\tfrac{5}{8} \text{ inch.}$$

II. Let $D=5\frac{1}{4}$ inches, $q=8$. How many teeth are there in the wheel, and what is its circular pitch?

$$\begin{aligned} N&=Dq \\ &=5\cdot 25\times 8=42=\text{No. of teeth.} \end{aligned}$$

$$P=\frac{\pi}{q}=\frac{3\cdot 1416}{8}=\cdot 393=\tfrac{13}{32} \text{ inch}=\text{pitch.}$$

241. Outline Drawing of Spur Wheels.—Having defined some of the parts of spur wheels, we will take an example of a drawing which shall include the parts considered.

Figs. 1 and 2, Plate IX., represent in elevation and plan a pair of spur wheels in gear; they are drawn in *outline*, that is, not showing the teeth. This is the usual way of representing toothed wheels in working drawings, for to take the time necessary to exhibit all the teeth is a useless expenditure of time and money. Fig. 2 shows half of each wheel in section.

The wheel on the axis A has twenty-four, and that on B has eighteen teeth, $\frac{3}{4}$ inch pitch. The pitch circles are marked P.C.; the dotted circle marked *t*, fig. 2, represents the *tops*, and that marked *b* the *bottoms* of the teeth (we shall refer to the teeth presently); A is a *plate* wheel, *c* is the plate which connects the boss *a* to the rim *d*. The wheel B is solid; the projecting pieces *e e* are termed *facings*. The figures are drawn to a scale of $\frac{1}{4}$.

To make our drawing we must first find the diameters of the pitch circles by equation (3), Art. 236, from which we can obtain the distance A B.

$$D=\frac{P}{\pi}\times N.$$

$$\left.\begin{array}{l}\text{For wheel A, } N=24\\ \text{For wheel B, } N=18\end{array}\right\}\ \frac{P}{\pi} \text{ for } \tfrac{3}{4} \text{ inch pitch}=\cdot 2386 \text{ (from Table XIII.)}$$

Substituting these values for wheel A in the equation, we get

$$D = \cdot 2386 \times 24 = 5 \cdot 726 = 5\tfrac{23}{32} \text{ inches nearly;}$$

for wheel B we get, $D = \cdot 2386 \times 18 = 4 \cdot 294 = 4\tfrac{9}{32}$ inches nearly.

$$\therefore \text{A B} = \frac{5 \cdot 726 + 4 \cdot 294}{2} = 5 \cdot 01 = 5 \text{ inches nearly.}$$

If A is the driver, the velocity-ratio $= \frac{CB}{AC} = \frac{18}{24} = \frac{3}{4}$.

242. Construction.—Draw the common central line A B, and mark off the distance AB just found; divide AB in C so that $\frac{AC}{CB} = \frac{24}{18}$, or take half the diameter already found as radii, and from A and B as centres describe the pitch circles. From C mark off along A B distances equal to the tops and bottoms of the teeth of each wheel, making the top $\frac{5\frac{1}{2}}{15}$, and the bottom $\frac{6\frac{1}{2}}{15}$, of the pitch; through these points describe the circles *t*, *b*, for each wheel. Other proportions of the teeth will be considered further on.

The remaining dimensions of the wheel A are as follows:—Thickness of rim *d*, $\frac{5}{8}$ inch; diameter of boss *a*, $2\frac{1}{4}$ inches; diameter of hole in boss for shaft, $1\frac{1}{4}$ inch; width of teeth, $1\frac{3}{4}$ inch; width through boss *a*, 2 inches; and thickness of plate *c*, $\frac{3}{8}$ inch. These dimensions are usually given in terms of the pitch, to which we shall refer later on. The wheel B is solid; the facings *e* are 2 inches diameter and $\frac{1}{8}$ inch thick; diameter of hole for shaft, 1 inch. These dimensions will enable the student to make a drawing of the wheels; there are, however, some points to which we shall have to refer presently.

243. Before considering the proper form of the teeth of wheels, we shall give an example of a spur wheel, showing a common way of drawing the teeth, as used for scale drawings; the method gives only an approximation to the true form, but for such purposes may be used with advantage. We shall give examples of a better method of approximating the true form of the teeth in a succeeding chapter.

244. We shall now give the proportions of the teeth, which are usually expressed in terms of the pitch. Fig. 234 shows a tooth divided into its several parts; P C is the pitch line, A D the pitch, T the top of the tooth, and B its bot-

tom. The top and bottom of a tooth are the portions respectively outside and inside the pitch line. The top of the tooth is sometimes termed the *face*, and also the *addendum*, while the bottom is called the *flank*. W is the thickness, T + B the total depth, and S the space between two consecutive teeth; A D, W, and S are measured along the pitch line; R is the thickness of the rim.

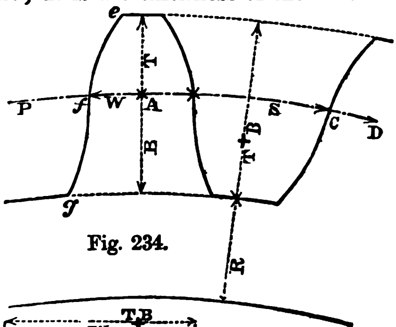


Fig. 234.

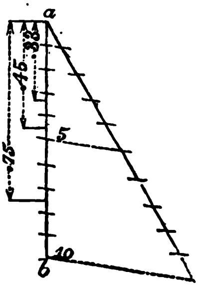


Fig. 235.

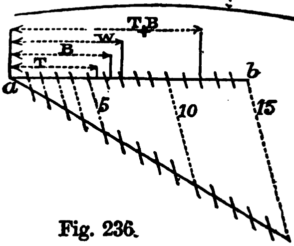


Fig. 236.

There are several sets of proprotions in use for different classes of work; we give three sets in general use. In the chapter on the teeth of wheels we shall show what determines the length of the teeth.

The set of proportions shown in fig. 236 has been used for the wheels in figs. 1, 2, 5, and 6, Plate IX., and is obtained by dividing the pitch into 15 equal parts, $a\,b$ = the pitch (= $1\frac{1}{4}$ inch in fig. 236)

$$T=\frac{5\frac{1}{2}}{15},\ B=\frac{6\frac{1}{2}}{15},\ W=\frac{7}{15},\ S=\frac{8}{15};\ \text{and } T+B=\frac{12}{15}.$$

$$\text{and the } \textit{play} \text{ or } \textit{back-lash},\ S-W=\frac{1}{15}.$$

In fig. 235 is shown another set of dimensions. *a b* is the pitch, as before, which is divided into 10 equal parts; of these

$$T=3{\cdot}3,\ B=4{\cdot}2,\ T+B=7{\cdot}5,\ W=4{\cdot}5,\ S=5{\cdot}5,\ R=4{\cdot}5;$$

or, calling the pitch p,

$$T=p\times{\cdot}33,\ T+B=p\times{\cdot}75,\ W=p\times{\cdot}45,\ S=p\times{\cdot}55,\ R=p\times{\cdot}45.$$

The width of the tooth depends upon the power to be transmitted; the usual width $=p\times 2{\cdot}5$.

The following proportions are given by Sir W. Fairbairn;* the letters refer to the parts in fig. 237, which is taken from the work named. In the right hand column is given the dimensions for a $2\frac{1}{2}$ inch pitch :—

		Proportional Part.		Inches.
Pitch,	=	1·00	=	$2\frac{1}{2}$
Depth,	=	0·75	=	$1\frac{7}{8}$
Working depth,	=	0·70	=	$1\frac{3}{4}$
Clearance,	=	0·05	=	$\frac{1}{8}$
Thickness,	=	0·45	=	$1\frac{1}{8}$
Width of space,	=	0·55	=	$1\frac{3}{8}$
Play or $fd-cf$,	=	0·10	=	$\frac{1}{4}$
Length beyond pitch line,	=	0·35	=	$\frac{7}{8}$

In fig. 237 is shown a scale of proportions of teeth, by means of which the proportions for any pitch from 0 to 4 inch may be obtained, and by increasing the number of inches, as seen on the left hand, those of any desired pitch may be found.

The distance on the horizontal line between 0 and 1″ may be any convenient length, but the line must be divided proportionally to the pitches. Thus, if the line is 4 inches long, and the greatest pitch is to be 3 inches, then divide the line into three equal parts to obtain the proportions for 1″ and 3″ pitches; and if fractions of these are required, divide the line accordingly. Make 0 B equal 3 inches, B *c*, B *d*, B *e* equal respectively the clearance, depth beyond pitch line, and depth within pitch line, etc., for a 3 inch tooth. Draw lines from *c*, *d*, *e*, etc., to 0.

245. Figs. 5 and 6, Plate IX.—These figures represent in elevation and plan respectively the spur wheel A, figs. 1 and

* *Treatise on Mills and Millwork*, Part II., page 33.

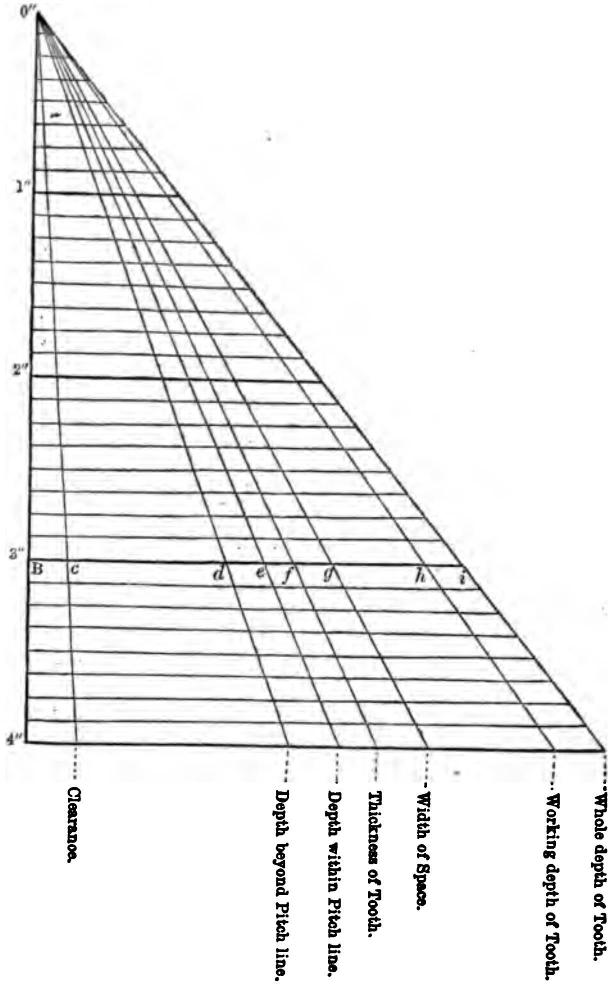

0"
1"
2"
3"
4"
B
c
d
e
f
g
h
i
Clearance.
Depth beyond Pitch line.
Depth within Pitch line.
Thickness of Tooth.
Width of Space.
Working depth of Tooth.
Whole depth of Tooth.

Fig. 237.

2, same Plate, the dimensions of which are given in Arts. 241 and 242, pages 196, 197, to a scale of $\frac{1}{2}$.

Having drawn the centre lines $e\,x$, $f\,y$ (the former contains the projections of the axis), the pitch circle S P, and the circles for the tops and the bottoms of the teeth, divide the pitch circle into twenty-four equal parts. Take one of the pitch points, as a, and mark on each side of it a distance $a\,b = \frac{1}{2}$ W; from d as a centre with a radius $d\,b$ (= the pitch + $\frac{1}{2}$ W) describe the top of the tooth $b\,b$; and from c as a centre with a radius $c\,b$ (= the pitch − $\frac{1}{2}$ W) describe the bottom of the tooth $b\,b'$ (the points d and c are the centres of the teeth on each side of a). Then $b\,b\,b'$ is one side of a tooth; by repeating the operation its other side can be drawn, and in like manner the remaining teeth of the wheel. The student will find it better first to draw all the tops and then the bottoms of the teeth, so that only one alteration of his drawing instrument will be necessary. Fig. 6 is an elevation, and fig. 5 is a plan, of the wheel; the right-hand half of the plan is in section, as made by a plane S_1P_1, fig. 6, showing the key in position; the other half of the plan is in ordinary projection showing the teeth; the construction lines indicate how each is obtained. The figures are drawn to a scale of $\frac{1}{2}$. We must remind the student that this way of drawing the teeth is only to be used in certain cases, as previously stated.

246. Rack and Pinion.—If we suppose the radius of one of a pair of spur wheels, as A in fig. 1, Plate IX., to become infinite, then the pitch circle would be a straight line, and the wheel would become a *rack*. Either the rack or the pinion may be, though generally the pinion is, the driver.

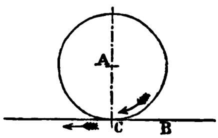


Fig. 238.

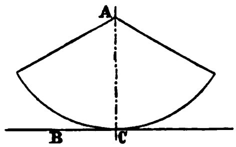


Fig. 239.

Fig. 238 shows one form of rack and pinion; A is the pinion, and B the rack; if the rotation of the pinion is

right-handed, the rack will move from right to left, as shown by the arrows; in fact, just as if the rack were an external wheel. Sometimes the pinion becomes large in diameter, and the extent of motion of the rack small, so that the pinion only turns through a fraction of its circumference, the pinion is then made in the form of a sector or segment, as shown in fig. 239. The rack may be a screw and the pinion a worm wheel; we shall refer to this presently. The linear velocity of the rack is equal to the perimetral velocity of the pitch circle of the pinion.

In Plate XII. are given three views of a rack and pinion in gear. Fig. 1 shows the plan of rack and pinion, fig. 2 the front elevation of the same, and fig. 3 an end elevation with the upper part of the pinion in section.

SECTION II.

BEVEL WHEELS.

247. If we wish to transmit rotary motion from one shaft to another with a constant directional relation and velocity-ratio, and the shafts are not parallel, we employ circular bevel wheels. There are two cases, firstly, where the shafts are in the same plane; and, secondly, where the shafts are not in the same plane. For simplicity we shall divide the first case into two: first, where the axes are at right angles; and second, where the axes are at an angle other than a right angle.

The bevel wheels in the second case are termed *skew-bevel* wheels; they are only employed in exceptional cases, owing to the difficulty of making accurate wheels; this reason, however, should not hold good, if in other respects it is advantageous to use them, as they can be readily constructed, though not so easily as ordinary bevel wheels. The same transmission of motion can be effected by a combination of ordinary bevel wheels, as we shall show presently.

We shall consider the wheels in the first instance to be toothless, so that the surfaces we represent are the *pitch-surfaces* of the toothed wheels; we treated spur wheels

exactly in the same manner. The pitch-surfaces of bevel wheels are frusta of cones.

248. Bevel Wheels having Axes at Right Angles.—Two axes A and B, whose projections are $a\,a$, $a'a'$, and b, $b'b'$, figs. 240 and 241, are to be connected by means of bevel wheels, so that the velocity-ratio shall be 2, A being the driver, that is, A is to make 2 revolutions to 1 of B.

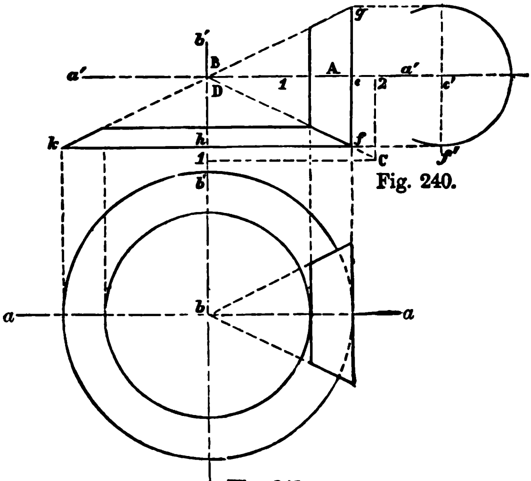


Fig. 240.

Fig. 241.

Let the two axes intersect in D, fig. 240; upon $b'b'$ set off from D any convenient length D 1 as a unit of length; and upon $a'a'$, from D, a distance D 2 equal to two of the same units of length. Upon D 1 and D 2, describe the rectangle D 1 C 2, and draw the diagonal D C. Let $e'f'$ be the greatest radius of the driving wheel, draw $f'k$ parallel to $a'a'$, meeting D C in h. Through f draw lines parallel to $a'a'$ and $b'b'$, meeting $a'a'$ in e, and $b'b'$ in h; make $e\,g = e\,f$, and $h\,k = h\,f$; then $g\,f$ and $k\,f$ will be the required greatest diameters of the wheel. Join D k, D g (k D g will be a straight line), then D $f\,g$, D $f\,k$ are two cones having a common vertex D, which, being centered upon the axes A and B ($a'a'$, $b'b'$), will revolve in contact, so that the axes A shall make two revolutions

while the axis B makes one. The line D f is called the *line of contact*; that is, the line in which the surfaces meet. Frusta of these cones are used for the wheels, as shown in the figures. Fig. 240 is an elevation, and fig. 241 a plan. We have stated with respect to spur wheels that the radii of the pitch circles of a pair of wheels in contact are inversely as their angular velocities, and from the above construction it will readily be seen the same statement holds with bevel wheels.

Let V and v be the respective angular velocities of A and B, $e f$ and $h f$ their radii, then

$$\frac{ef}{hf} = \frac{v}{\text{V}} = \tfrac{1}{2}.$$

The velocity-ratio of

$$\frac{\text{A}}{\text{B}} = \frac{hf}{ef} = \frac{\text{V}}{v} = \frac{2}{1} = 2.$$

249. When the wheels are of equal diameter, and the axes at right angles, they are called *mitre bevel wheels*. Figs. 3 and 4, Plate IX., represent a pair of such wheels in gear; we shall refer to this example shortly.

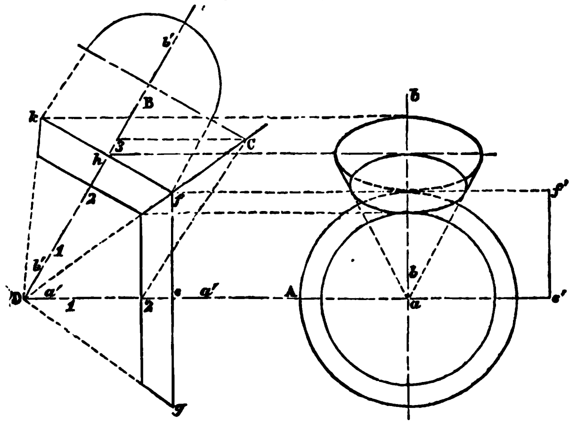


Fig. 242. Fig. 243.

250. Bevel Wheels having Axes not at Right Angles. —Two axes A and B, whose projections are a, $a'a'$, and $b\,b$,

$b'b'$, figs. 242 and 243, are inclined to each other at an angle of 60°, and lie in the same plane. They are to be connected by means of bevel wheels, so that the velocity-ratio shall be $\frac{2}{3}$; the greatest diameter of the wheel on A, which is the driver, to be equal to twice $e'f'$, fig. 243.

Draw the axes b'D and D a', fig. 242, to contain an angle of 60°, and let them meet in the point D. Upon D a' set off D 2 = 2 units of length (D 1), and upon D b' set off D 3 = 3 of the same units. Upon D 3 and D 2, describe a parallelogram D 2 C 3, and draw the diagonal D C, which is the line of contact. Draw ff' parallel to $a'a'$, meeting D C in f; through f draw $f e g$, $f h k$ perpendicular to $a'a'$, $b'b'$, respectively, meeting $a'a'$ in e, and $b'b'$ in h. Make $e g = e f$, and $h k = h f$; join D k, D g, then Dfk, Dfg are two cones having a common vertex D, which are the pitch surfaces of the required wheels. Frusta of these cones are used for the pitch surfaces of the wheels; we will explain more fully the form of the pitch and other surfaces in the examples that follow.

The velocity-ratio of

$$\frac{A}{B} = \frac{hf}{ef} = \frac{2}{3}.$$

251. Mitre Bevel Wheels.—We have already explained that mitre bevel wheels are those whose axes are at right angles, and diameters equal; we now proceed to work out an example.

Figs. 3 and 4, Plate IX., are elevation and plan of a pair of mitre bevel wheels in gear; they are drawn in outline, and half of each wheel in fig. 3 is in section. The directional relation and velocity-ratio are constant, and as the radii of the wheels are equal, the velocity-ratio is unity. The formula previously given for toothed wheels apply equally to bevel wheels.

If each wheel has 24 teeth, 1 inch pitch, therefore the diameter of the pitch circle $= \frac{P}{\pi} \times N = \cdot 3182 \times 24 = 7\cdot 636$, or $7\frac{5}{8}$ inches nearly.

252. Draw the projections $a\,a$, $a'\,a'$, and $b\,b$, b' of the axes, and let C, fig. 3, be their point of intersection. From b', fig. 4, as a centre, describe the pitch circle of the wheel B, and

draw the pitch line of the other wheel at right angles to $a'a'$, and tangential to the pitch circle already drawn. Now draw $g\,f$, $f\,k$ the plans of the pitch circles, which may be done by describing from C as a centre, a circle equal in diameter to the pitch circle, and drawing $g\,f$ and $f\,k$ at right angles to the axes and tangential to the circle. Join Cf, Cg, and Ck, g C k will be a straight line, then C$g\,f$ and C$f\,k$ are the conical pitch surfaces of the wheels, and Cf the line of contact. Upon g C or f C mark off $g\,m$ equal to the width of the pitch surface, that is, equal to the width of the teeth; from m draw a line meeting fC, and cutting off a frustum of the cone f C g; then the frustum thus cut off is the whole pitch surface of the wheel A, and B is similar to it. The tops of the teeth are formed outside, and the bottoms inside this pitch surface.

From f draw $f\,e$ at right angles to the line of contact Cf, meeting the axis $a\,a$ in e, and join $e\,g$; then $e\,g$ is at right angles to Cg. Draw similar lines for the other wheel. From f, along $b\,f\,e$, set off the top t and the bottom h of the teeth of each wheel, as shown for clearness at g; from each of these points draw lines to C, and lines parallel to $g\,f$, $f\,k$. Complete the figure, as shown by the construction lines, and from the following dimensions:—Diameter of boss $2\frac{3}{4}$ inches, diameter of hole for shaft $1\frac{1}{2}$ inch, and width through boss $2\frac{5}{8}$ inches; width of teeth $2\frac{1}{2}$ inches. The other proportions of the teeth are to be taken from one of the sets of dimensions given in Art. 244, page 199. The teeth of bevel wheels are made of the same size at $t\,g\,h$, as the teeth of spur wheels of the same pitch, but as they radiate to a common centre C, they decrease in size the nearer they are to that centre, so that we have a maximum and a minimum size of tooth, and in calculating the strength we must take the mean of these.

253. The true form of the surface upon which the teeth are set out, as shown at $t\,g\,h$, is spherical; but as the surface occupied by the teeth is relatively small, they are set out on the surface of the *tangent cone* to the spherical surface at g. In fig. 7, Plate IX., we have shown the wheels circumscribed by the sphere of which they are parts, and the tangent cones; also the inscribed sphere and its tangent cones, upon which are set-out the minimum size of the teeth. In fig. 8 is shown

a portion of the wheel and the circumscribed sphere, drawn full size. The same letters of reference are used for all the figures. We shall have occasion to refer to this point when we treat of the teeth of bevel wheels.

254. Pair of Bevel Wheels in Gear.—We now give another and a more general example of a pair of bevel wheels, drawn in outline; the axes are at right angles, and the velocity-ratio $\frac{5}{6}$. The driver has 24, and the follower 20 teeth, $1\frac{1}{2}$-inch pitch. Figs. 1 and 2, Plate X.—A is the driver and B the follower; the projections of their axes are marked *a a*, *a'*, and *b b*, *b' b'*. If R, *r* are the radii of the pitch circles, and N, *n* the numbers of teeth in A and B respectively, then

$$\frac{R}{r} = \frac{N}{n}. \text{ The velocity-ratio} = \frac{r}{R} = \tfrac{5}{6}.$$

Find the diameters of the wheels by the equation previously given (page 193); thus, for wheel A, $D = \frac{P}{\pi} \times N = \cdot 4774 \times 24 = 11\cdot 457 = 11\frac{29}{64}$ inches nearly; for wheel B, $D = \cdot 4774 \times 20 = 9\cdot 548 = 9\frac{33}{64}$ inches nearly.

255. Draw the centre lines *a x*, *b b*, and *b' b'*, and the projections of the pitch circles, as shown; let C be the point of intersection of *a a* and *b b*. From *f* draw *l f* at right angles to C *f*, meeting *a a* in *e*, and *b b* in *l*. From *e* draw *e g*, and from *l* draw *l k*, and produce them, making *g t*, *k t* equal in length to the top of a tooth from one of the sets of proportions given previously. Then *t e t* and *t l t* are the cones upon the surface of which the teeth are to be set-out; make *g h* equal to the outside of a tooth, and draw the lines *t t'*, etc., which represent the tops and the bottoms of the teeth. Make *g m* equal to the width of the teeth; from *m* draw *m n* parallel to *g e*, from *n* draw *n o* parallel to *e l*, and from *o* draw *o p* parallel to *l k*; produce each of these lines to meet lines drawn from *t t*, and *h* in *t' t'* and *h'*. Then *t' n t'* and *t' o t'* are the cones upon the surfaces of which the inside, calling *t g h* the outside, of the teeth are set-out.

Fig. 3 shows the developments of the cones *t e t* and *t l t*; only a portion of the latter is shown. Upon these developments the teeth are set-out in their true size, as it will readily be seen they are not shown so in the projection, fig. 1. As the surfaces of the teeth radiate to a common centre C, it is

not necessary to draw the developments of the smaller cones. The line E L is drawn parallel to *e l*, *f* F is C *f* produced, and *e* E, *l* L are drawn parallel to *f* F.

The dimensions of the parts not already given may be taken from the figures and read off on the scale attached.

SECTION III.

SCREW OR HELICAL-TOOTHED WHEELS—SPUR MITRE WHEEL—WORM AND WORM WHEEL.

256. There is a class of wheels often employed to connect two shafts, called screw or helical-toothed wheels. These wheels take certain distinct forms, according to the position of the axes and of the relative number of threads or teeth on them. The general term, screw wheels, includes them all, but for convenience it is advisable to divide them into three classes; we therefore arrange them as follows:—

CLASS I.—Axes parallel and in the same plane.
,, II.—Axes in parallel planes, and their projections on either plane at right angles.
,, III.—Axes in parallel planes, and their projections on either plane forming an angle other than a right angle.

The wheels in Class I. are seldom employed on account of the increased friction between the teeth, due to their lateral oblique action, and for the majority of purposes ordinary spur wheels can be used instead; the latter are also easier to make, and work with much less friction. We shall therefore only briefly refer to this class of wheels. The wheels of Class II. are extensively employed, and it is convenient to divide them into two sections; first, those in which the number of teeth in each of the two wheels are either equal, or do not vary beyond a limit of about—for there is no exact line to the limit—four or five times as many in the one as there are in the other. Second, those in which the velocity-ratio of the two connected shafts is greater than five, being generally twenty or upwards. The wheels of Section I. are called screw or helical-toothed, and in the case of equal wheels they are sometimes called spur mitre wheels. A pair of those forming

the second division are called a worm and worm wheel, the smaller, or pinion, is termed the worm or endless screw. We shall give examples of these wheels.

The wheels of Class III. are seldom employed, we therefore merely state that the rules given for those of Class II. can be applied, with a little variation in detail, to those of Class III.

257. All screw wheels work more or less by friction, and this friction is due to the oblique lateral pressure exerted between their acting surfaces, which are inclined to their axes of rotation; in other words, helical teeth have a tendency to increase friction, owing to the lateral oblique pressure they exert; but they work by rolling and not by sliding contact. A portion of this laterally oblique pressure is thrown upon the bearings that carry these axes, so that besides the friction of the acting surfaces of the teeth, there is that thrown upon the bearings, in the form of end pressure, between the collars on the shafts and the faces of their bearings. This increase of friction is objectionable, as it is a waste of power, and hence this class of wheel work is only employed in cases where it cannot be dispensed with, for want of space, etc., or in those cases where the advantage gained more than compensates for the loss due to increased friction.

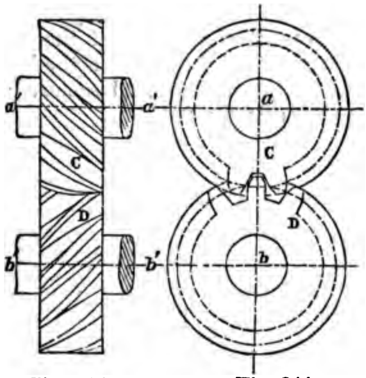


Fig. 245. Fig. 244.

258. Class I.—*Axes parallel and in the same plane.*—In figs. 244, 245 is shown a pair of screw or helical-toothed wheels

in gear, connecting the shafts A and B, whose projections are $a, a' a'$, and $b, b' b'$. The wheel on the shaft A is marked C, and that on the shaft B, D. These wheels resemble spur wheels, but instead of having their teeth at right angles with their faces, or parallel to their axes, they are inclined to them. Such wheels may be considered as portions of screws having a number of threads upon them, the number of threads answering to the number of teeth. For a pair of such wheels to gear correctly, the following conditions must be fulfilled:—If they are external wheels, as in figs. 244 and 245, the screw-threads must be right-handed on one and left-handed on the other; if one of the two is an internal wheel, then they must be either both right-handed or both left-handed. The teeth must be inclined to the axis of each wheel at the same angle; that is, the developed angle or inclination, or obliquity of the teeth, must be the same in each wheel. The circular pitch, that is, the distance from centre to centre, or from outside to outside, of two adjacent teeth or threads on the pitch circle, as seen in the front elevation, fig. 244, must be the same in each wheel.

These wheels were invented by Dr. Hooke, the inventor of the stepped tooth, Hook's joint, etc. If the angular spaces formed by the steps K L and M, etc., fig. 289, were filled up to a line drawn to touch the corners of the stepped teeth, as the corners *b*, *d*, and *f*, of the teeth on the left hand of K L and M, the tooth thus formed would become a helical or screw tooth, as those shown in fig. 245.

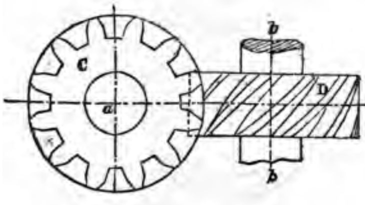


Fig. 246.

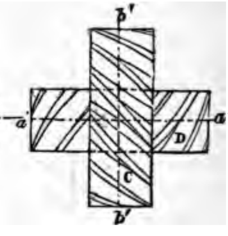


Fig. 247.

258. CLASS II.—*Axes in parallel planes, and their projections on either plane at right angles.*—Before giving a general

solution, we will take the special case of equal wheels, or as they are called, spur mitre wheels. In figs. 246 and 247 are shown a pair of equal screw wheels C and D, with their axes A and B, whose projections are a, $a'a'$ and $b\ b$, and $b'b'$, each wheel has twelve teeth or threads, which are right-handed screw-threads. The wheels are in fact portions of a twelve-threaded right-handed screw, of such a pitch that the sum of the angles of inclination of the two threads, one in each wheel, are equal to a right angle, which is the angle contained by the projections $a'a'$ and $b'b'$ of the axes, fig. 247. These wheels are convenient for transmitting motion from one shaft to another, where the distance between them is relatively small, but the objection exists that they work by friction; however, this is more than compensated for in cases where they are employed by the advantage gained. Cast steel has been found to answer well as a metal for such wheels. Messrs. Collier, and Sir J. Whitworth, of Manchester, both employ screw wheels for driving the drill spindles of their multiple drilling machines. As the wheels are equal and similar, there is no change in the velocity of the shafts A and B. The case of unequal wheels, and of wheels of different numbers of threads, will be considered in the next article.

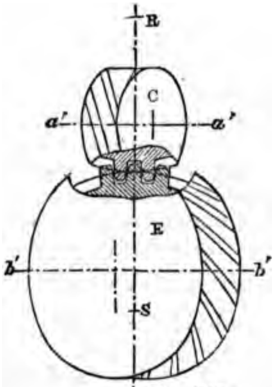


Fig. 249.

259. Class II.—*General Case.*—In figs. 248 and 249 are shown a pair of screw wheels C and E of unequal diameter, and having an unequal number of teeth. The projection of the axes are, as in the former case, at right angles, but by a slight change, the construction we shall give may be applied to cases where those projections are not at

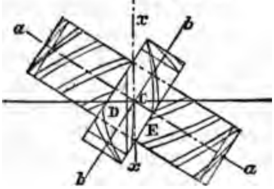


Fig. 248.

right angles. The projections of the axes A and B are marked $a\,a$, $a'a'$, $b\,b$, $b'b'$. Unlike spur or bevel wheels, the velocity-ratio of the axes connected by screw wheels is not independent of the radii of their pitch surfaces. The plan, fig. 248, is inclined to the plane of projection of fig. 249. The portion of fig. 249 in section is made by a plane x C x at right angles to the plane of projection of fig. 249.

The angular velocity-ratio of a pair of screw wheels in gear depends upon the number of threads or teeth in each, and are inversely as those numbers. These wheels under consideration have six and eighteen teeth, and the radii of their pitch surfaces, which are cylindrical, are as 1 : 2. The velocity-ratio, C being the driver, of

$$\frac{C}{D} = \frac{18}{6} = \frac{3}{1}.$$

That is, C makes 3 revolutions to 1 of D.

For continuous motion to be transmitted by these wheels, the velocity-ratio of a pair of wheels in gear must be expressed by a whole number, just as in the case of spur wheels, for the circular elevation of a screw wheel resembles a spur wheel, as may be seen in fig. 246, and in the examples to follow, especially those illustrated by Plates XXI. and XXII. In other words, the pitch, as measured on the pitch circle, must be an exact divisor of that pitch circle; thus, if there are six teeth, then the pitch $= \frac{\text{circumference of pitch circle}}{6}$, This pitch is the divided *circular pitch*, and it is generally different in each wheel of a pair.

The divided normal pitch of two wheels in gear must be the same. If the screw threads of a pair of wheels in gear are either both right or both left handed, then the sum of the angles of inclination of two threads in contact, one on each wheel, must equal the angle contained by the projection of their axes; that is, if the axes are at right angles, and one makes an angle of 60° with its axis, that of the other must make an angle of 30°. If one is right-handed and the other left-handed, then the angle between the axes must be equal to the difference of the angles of inclination. In figs. 250 and 251, A is shown as a cylinder, the projections of whose axis are a, $a'a'$, which we shall consider as the pitch

surface of a screw wheel on which is traced one revolution, turn, or coil, of a helix, 0...XII in fig. 250, and 0...12 in fig. 251. The square 0, 12, XII B, fig. 252, is the development of the cylindrical surface, and the line 0...XII the development of the helix. The line 0...12 is equal in length to the circumference of the cylinder, and the line 12...XII

Fig. 250.

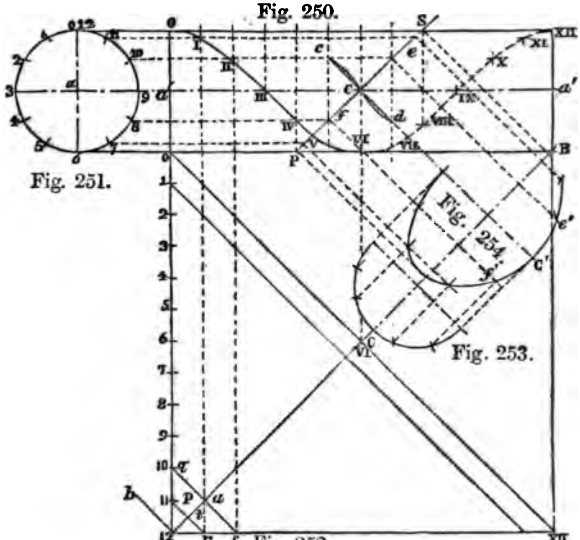


Fig. 251.

Fig. 252.

Fig. 253.

Fig. 254.

is equal to the pitch of the helix; that is, the distance measured on the surface of the cylinder, and in a direction parallel with its axis, between two corresponding points of one coil of the helix. This pitch, for distinction, is called the *axial pitch*, it is the ordinary pitch of a screw. If from the point 12, fig. 252, we draw a line, 12...VI, at right angles to the line 0...XII, that line represents a portion of the development of the normal helix; that is, a helix traced upon the same cylinder, but which cuts the first helix at right angles. If the cylinder were produced towards the left and the helix continued, another coil would commence at the point 12, as 12 *b*, in the development, fig. 252; so that the

line 12...VI is the shortest distance between two coils of the helix. At C, fig. 250, is shown a portion of the first helix *c* C *d*, and a portion of the normal helix, marked *e* C *f*. The line 12...C, fig. 252, is called the normal pitch of the helix 0...XII, and as before stated, the line 12...XII (or O B), is the axial pitch. The line 0...12 may be called the *circular pitch*, for it is the length of the circumference of the cylindrical surface which forms the pitch surface upon which the helix is traced, and in the development it forms one side of the triangle 0, 12, XII.

Let the circumference or circular pitch, 0...12 = g, the axial pitch 12...XII = h, and the development of one coil of the helix 0...XII = k, then

$$k=\sqrt{(g^2+h^2)};$$

g, h, and k being expressed in inches and fractions of an inch, or in any other convenient notation.

Now, suppose instead of there being only one helix 0...XII traced upon the cylinder, there are a number of smaller helices, say twelve, and that they are all at an equal distance apart, in fact, in the end elevation, fig. 251, they divide the circle 0...6...12 into twelve equal arcs; and suppose these twelve helices are traced upon the cylinder and a development of the whole made, then we should have, in the present case, supposing the cylinder is of the length shown in fig. 250, a square, fig. 252, with a number of lines 11*r*, 10*s*, etc., drawn parallel to the developed helix 0...XII, which is in this case a diagonal of the square, these developed helices between 0...XII and 11*r* would divide the lines 0...12, 12...C, and 12...XII. In fig. 252 we have divided the circumference 0...12 into the same number of equal parts as on the circle, fig. 251; from these points 0...10, 11 lines are drawn parallel to 0...XII cutting the line 0...VI, the normal pitch, in *t*, *u*, etc., and the line 12...XII the axial pitch in *r*, *s*, etc. We thus divide the three pitches, circular, normal, and axial, into as many parts as there are helices on the surface, and instead of speaking of the whole or total pitch of the helix, we refer to the divided pitch; thus the line 10, 11 is the *divided circular pitch*; the line *t u*, the *divided normal pitch*; and the line *r s* the *divided axial pitch*. The length of each

of these divided pitches is found by dividing each of the total pitches by the number of helices; in the first case there are twelve, thus the divided normal pitch, for example, $= \frac{1}{12}$ of the line 12...VI.

260. Radius of Curvature.—If we take a section of the cylinder, fig. 250, at right angles to the helix 0...XII, we shall have for that section an ellipse, a portion of which is shown in fig. 254. This section is made by a plane S P, at right angles to the direction of the portion of the helix *c d* at the point C; the point C′, fig. 254, is a projection of C, and *e′f′* are projections of *e* and *f*. The radius of a circle which corresponds to a small arc of the ellipse at the point C′, is called the radius of curvature at that point; practically, the arc *e′*C′*f′* corresponds to the arc of a circle whose radius is equal to the radius of curvature for the point C′. A small portion of the normal helix *e f* on each side of the point C, fig. 250, lies in the section plane S P, so that the radius of curvature of the ellipse at the point C is also the radius of curvature of the normal helix at that point. In the present case, the pitch of the helix is equal to the circumference of the cylinder on which it is traced; that is, the axial and circular pitches are equal, therefore the development of 0, 12, XII, B of the cylinder is a square, and that of the coil 0, XII, a diagonal of that square. Therefore the original helix and the normal helix, portions *c d* and *e f* of which are shown intersecting at C, fig. 250, are each inclined to the axis of the cylinder at an angle of 45°; and the radius of curvature of the normal helix equals the diameter of the cylinder.

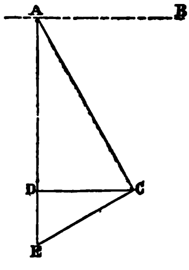


Fig. 255.

In fig. 255 is shown a construction for obtaining the radius of curvature of a helix. Let A B represent the axis of the cylinder upon which the helix is traced. Draw A D at right angles to A B and equal to the radius of the cylinder; make the angle B A C equal to the angle of inclination of the screw. From D draw D C perpendicular to A D, meeting A C in C, and from C draw C E perpendi-

cular to A C, meeting A D produced in E, then A E is the radius of curvature.

By calculation $A\,E = A\,D\left(1 + \frac{p^2}{c^2}\right)$; where p stands for the axial pitch of the helix, and c for the circumference of the cylinder; p and c we have also called the axial pitch and the circular pitch, respectively.

261. Worm and Worm Wheel.—When the velocity-ratio to be transmitted between two shafts at right angles is great, as for example when one shaft is to make, say, 25 or 50 revolutions to one of the other, a worm and worm wheel is usually employed. These are screw wheels; but, as we shall see, owing to the extreme care required, the teeth are usually constructed upon different principles, and more resemble those of a rack and spur wheel. The smaller wheel is called a worm or endless screw, and is in fact a portion of a screw, while the larger is called the worm wheel, and in some cases resembles a spur wheel with its teeth set at an angle with the axis, and not parallel to it, as in the case of an ordinary spur wheel.

In figs. 256 and 257 is shown a worm and worm wheel in gear; the threads of the worm and the teeth of the wheel are both right-handed. The worm D is the driver and the wheel E the follower; the projection of these axes A and B are $a\,a$, a', b, $b'b'$. The velocity-ratio of the shafts A and B depends upon the number of threads in the two wheels, and is independent of the diameters of their pitch surfaces; in this respect they resemble screw wheels.

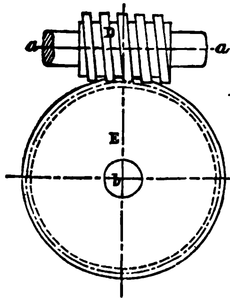


Fig. 256.

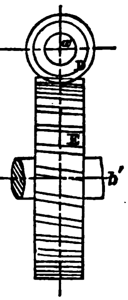


Fig. 257.

If the worm is a single-threaded screw, which is usually the case, then for every revolution it makes, a radius of the worm wheel will describe an angle which is subtended on its

pitch circle by an arc equal to the circular pitch of the worm-wheel, and as this arc is contained a whole number of times in the pitch circle the angle can be easily found. Thus, let there be 50 teeth in the wheel; then for every revolution of the worm, the wheel makes $\frac{1}{50}$ of a revolution; and the angle described by the radius $= \frac{360}{50} = 7{\cdot}2°$. The velocity-ratio $\frac{A}{B} = \frac{50}{1} = 50$.

In fact, if we consider the worm as a wheel of one tooth, then we may express the velocity-ratio just the same as in the case of ordinary spur wheels; and if the worm is double-threaded, then we may consider it a wheel of two teeth, and so on. The *pitch* of a worm and worm wheel is usually understood as the *circular pitch* of the wheel and the *axial pitch* of the screws; but if we are considering a double-threaded screw, then it is necessary to define the term more exactly, otherwise error may result. We shall therefore treat the pitch of worms and worm wheels in a similar manner to what we have done screw wheels, which they in many respects resemble.

The *circular pitch* of a worm and worm wheel—

$$= \frac{\text{Circumference of the pitch circle}}{\text{Number of teeth}}.$$

The *normal pitch* is found in the same manner as for a pair of screw wheels, Art. 259, page 212; we shall not require to refer to this pitch in the case of worms and worm wheels. The *axial pitch* is the same as in the case of screw wheels, and is found by dividing the axial pitch of the helix by the number of teeth. The axial pitch of the worm is equal to the circular pitch of the worm wheel; these are the quantities usually required. The form of the teeth is considered in a previous article.

262. Skew Bevel Wheels.—Skew bevel wheels, as previously intimated, are employed to connect two shafts that are not in the same plane. The figures on the next page (258 and 259) represent the pitch surfaces of a pair of skew bevel wheels *in gear*, their pitch surfaces are *hyperboloids*, as shown by the dotted lines in fig. 258, and partly dotted lines in both figures; now as the portions of the hyperboloids used for the wheels

are small, tangent-cones to the hyperboloids are used for the pitch surfaces. The case we propose to illustrate is that in which the axes are in parallel planes, and the projections of those axes upon either plane are lines which are not parallel. The upper figure (259) is a projection upon a plane at right angles to the parallel planes, containing the axes A and B, whose proportions are marked *a′a′*, *b′b′*. The lower figure (258) is a projection upon a plane parallel to the planes containing the axes, and therefore at right angles to the plane of projection of the upper figure; the projection of the axes are marked *a a* and *b b*.

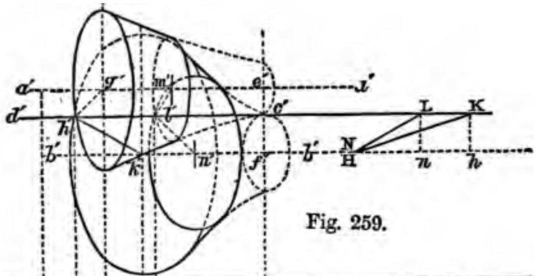


Fig. 259.

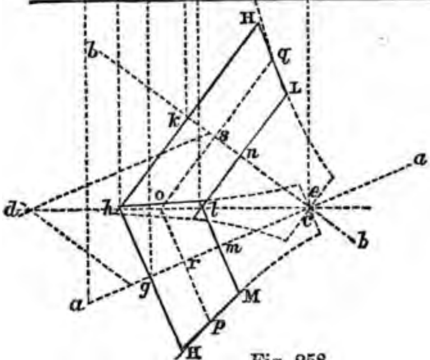


Fig. 258.

The perpendicular distance between *a′a′* and *b′b′* is the distance between the planes containing the axes, and *e′f′* is equal to that distance, which is the shortest between the two

axes. The angle $a\,c\,b$, fig. 258, is the *projected angle* contained by the axes, and it corresponds to that contained by the axes of a pair of ordinary bevel wheels. The two pitch surfaces meet in a line C D, called the line of contact, whose projections are $c\,d$ and $c'd'$.

263. Projection of a Pair of Skew Bevel Wheels—Let the wheel on the axis A be the driver, the velocity-ratio being $3:2$ or $\frac{3}{2}$, the angle $a\,c\,b = 60°$; the distance between the planes containing the axis $= e'f'$, figs. 258, 259; also, let the distance $e\,g$ and the width $h\,l$, measured on the line of contact $c\,d$, be given. Draw $a\,a$, $b\,b$, $a'a'$, and $b'b'$, let $a\,a$ and $b\,b$ intersect in c; then from c draw $c\,d$, as if $a\,a$ and $b\,b$ were the axes of a pair of ordinary bevel wheels. Divide $e'f'$ in c', so that $f'c' : c'e' :: 3:2$; through c', parallel to $a'a'$, draw $c'd'$, which is a projection of the line of contact of the two hyperboloids.

Through g draw $g\,h$ at right angles to $a\,a$, meeting $c\,d$ in h, and from h draw $h\,k$ at right angles to and meeting $b\,b$ in k. If the wheels were ordinary bevel wheels, $g\,h$ and $k\,h$, assuming these lines to be in the same plane, would be the greatest radii of the conical pitch surfaces; but as the axes are not in the same plane it is clear the radii in question must be greater than $g\,h$, $k\,h$. The projections of $g\,h$, $k\,h$, in fig. 259, are marked $g'h'$, $k'h'$, and h' is the point in which these lines meet. The true lengths of the lines, whose projections are $g\,h$, $g'h'$, and $k\,h$, $k'h'$, are the greatest radii of the pitch surfaces; these surfaces being tangent cones to the hyperboloids at p and q.

To find p and q mark off $h\,l$, and bisect it in o; through l draw lines at right angles to $a\,a$, $b\,b$, meeting them in $m\,n$, then $g\,m$ is the thickness of the frustum of the wheel on A, and $k\,n$ the thickness of that on B; p and q are the mean radii of these frusta, which can be found by a similar construction to that employed for the extreme radii, as follows:—

To find the lengths of the greatest radii G H, K H, and also of the least radii M L, N L, we will take those for the larger wheel B. Take a right-angled triangle H h K, shown on the right of fig. 259, H h being equal to $h\,k$, and h K to $f'c'$; then K H is the required greatest radius. The right-angled triangle N n L has its sides N n, L n equal respectively $n\,l$, $f'c'$; then N L is the least radius. In a similar manner

the radii of the smaller wheel may be found, the common side for the triangles being equal to $c'e'$, and their bases to $g\,h$ and $m\,l$.

Make the radii k H, n L, m M, and $g\,h$, fig. 258, equal to the greatest and least radii of the frusta, and join H L, H M. Then H $k\,n$ L, H $g\,m$ M, are the half conical frusta of the required pitch surfaces; the remaining halves, which are equal and similar, may easily be drawn.

From the above construction it is perfectly obvious that if the teeth of a pair of skew bevel wheels be cut in the direction of the generating line of the two rolling hyperboloids, that they will accurately meet, for this line is the line of contact of the two surfaces. For actual working, narrow frusta only are employed.

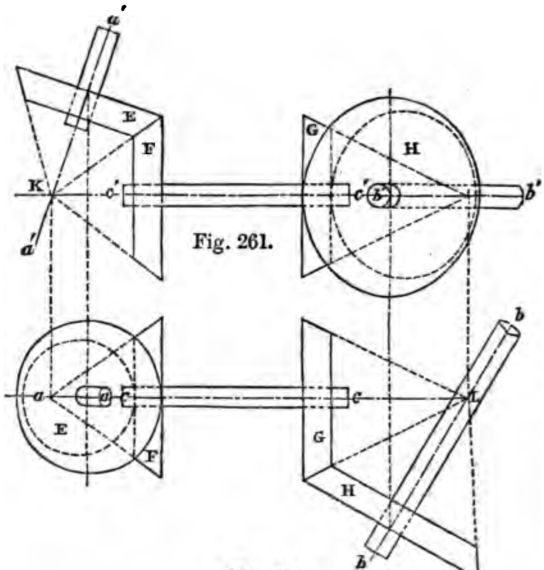


Fig. 261.

Fig. 260.

264. Combination of Bevel Wheels.—A more general arrangement for transmitting motion from one shaft to

another, in which the constant velocity-ratio of those axes is maintained, and in which the axes may have any possible relative position, is shown in figs. 260 and 261. The arrangement consists simply of two pairs of ordinary bevel wheels.

The two axes A and B, whose projections are $a\,a$, $a'a'$, and $b\,b$, $b'b'$, are provided with wheels E and H, which, supposing it were possible for them to gear together, would maintain the required velocity-ratio; but as they cannot gear together, an intermediate axis C, whose projections are $c\,c$, $c'c'$, is employed, upon which is fixed two wheels F and G, whose pitch circles are of equal diameter, these wheels being in gear with E and H respectively.

We have already shown that by means of a pair of bevel wheels we can connect two axes which are inclined to each other, and which lie in the same plane; in the present case we have the axes A and C in one plane, and $a'\,\mathrm{K}\,c'$ is the angle contained by them. The axes B and C are also in one plane, but not in the same one as A and C, and $b\,\mathrm{L}\,c$ is the angle contained by them. It is therefore clear that the arrangement, as stated, is possible whatever be the relative position of the axes A and B. The distance between the wheels F and G may be varied according to circumstances.

Let e, f, g, and h denote the radii of the pitch circles of E, F, G, and H respectively, F and G being equal. Then the velocity-ratio of

$$\frac{\mathrm{A}}{\mathrm{C}}=\frac{f}{e}, \text{ and of } \frac{\mathrm{C}}{\mathrm{B}}=\frac{h}{g};$$

therefore the velocity-ratio of

$$\frac{\mathrm{A}}{\mathrm{B}}=\frac{h}{e},$$

which is the same as if the wheels E and H were in gear.

SECTION IV.

ECCENTRIC WHEELS—ELLIPTIC WHEELS, ETC.

265. In the previous sections of this chapter we have treated of circular wheels, the axes of which have a constant velocity-ratio; we now propose to consider some of the more common forms of wheels which are not circular, but whose axes are parallel. Wheels of this class are only used for special purposes, but they are of sufficient importance to demand a space in this work. All wheels that rotate or oscillate about a fixed centre, whatever be the form of their outline or pitch surface, if their radii vary, may be called non-circular wheels. Thus an eccentric circular wheel, that is one whose pitch surface is circular, but whose centre of motion is not its geometrical centre, may be called a non-circular wheel, because all the radii of each similar half vary. However, in the present classification, we shall consider circular eccentric wheels separately.

In circular eccentric wheels, and in non-circular or rolling curve wheels, with parallel axes, the velocity-ratio is variable, and in the cases we shall consider the directional relation will be constant.

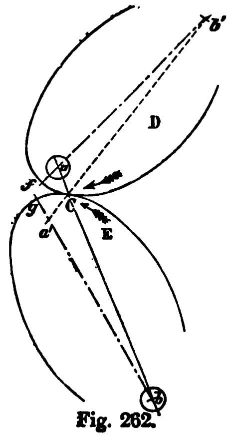


Fig. 262.

266. General Principles. — The fundamental condition that must exist in all cases of the transmission of motion by wheel work in gear, is that for every possible position of the two wheels while in gear, the sum of the radii must be constant for every one of those positions, these radii being one in each wheel, and their sum equal to the line of centres with which they coincide when their extremities are in contact. Thus, in fig. 262, let a and b be the centres of the projection of the parallel axes of a pair of rolling curve wheels, $a\,b$ the line of centres, and C in $a\,b$ the point of contact of the traces of the pitch surfaces, and let D be the

driver, rotating as shown by the arrow. The line of contact is a line passing through C perpendicular to the plane of projection. Suppose the point f on the wheel D, and the point g on E coincided at a given instant with C; then supposing there to have been no sliding during the change of position of these points, the arc C f would be equal to the arc C g. This condition is essential, as is also the following:—Let $a\,f$, a C, and $b\,g$, b C be the radii before and after the assumed rotation of D and E respectively. Then—

$$af+bg=aC+bC=ab,$$

all the points being in the plane of projection, which is perpendicular to the axes. The angular velocity-ratio of the axes of D and E, for their position shown in the figure—that is, with C for the point of contact—is the same as if the wheels were circular, that is, the angular velocity-ratio of $\frac{D}{E}=\frac{bC}{aC}$.

But this ratio varies for every new position of the wheels, it increases or decreases according as the radius of D decreases or increases. In the figure, the two wheels D and E are portions of equal and similar ellipses entered at their foci a and b, the foci a' and b' being the revolving foci.

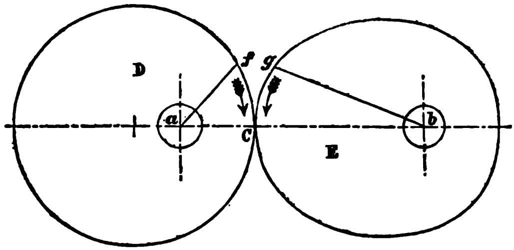


Fig. 263.

267. Eccentric Wheels.—An eccentric wheel is one whose centre of motion, as the centre of its axis, is not the geometrical centre of the trace of its pitch surface; this trace may be a circle or some other curved line, as, for example, in the above illustration. In fig. 263 is represented a pair of wheels D and E fixed on axes A and B, whose projections are a and b; $a\,b$

is the line of centres, and C the point of contact, the line of contact is perpendicular to the plane of projection of the figure, and passes through C. The wheel D is a circular eccentric wheel, and the wheel E is a non-circular eccentric wheel. The circumference of the trace of the pitch surface of the wheel E is equal to that of D, and the sum of the radii of the two wheels for each point of contact is constant, and is equal to the line of centres $a\,b$. Thus, suppose $a\,f$ and $b\,g$ are two radii, one in each wheel, then for the wheels to gear correctly we must have the arc $C\,g =$ arc $C\,f$, and $b\,g + a\,f = a\,b$; and so on for each new position of the two points f and g which are to come into contact. In designing such a pair of wheels, the circular one D is first drawn, and then the non-circular one E is drawn to work with it, according to the conditions before stated. This class of wheelwork is only used for special purposes where a varying velocity-ratio is required.

An example of non-circular wheels is given in this and the next sections. In the present case we have an elliptical wheel, centered at its geometrical centre, that is at the point of intersection of its major and minor axes in gear with an eccentric circular wheel.

Elliptic wheels centered at their foci are a common form of eccentric wheels.

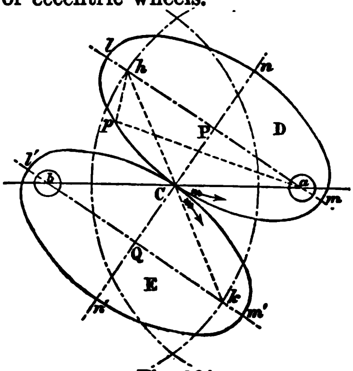


Fig. 264.

All eccentric wheels require to be balanced if they rotate at a high rate, or are comparatively heavy, as they are not self-balanced as circular wheels are.

268. Elliptic Wheels.—Two equal and similar elliptic wheels centered at their foci, are rolling curves which fulfil the conditions stated in Art. 266.

In fig. 264 is shown two such ellipses, D and E, fixed on axes, A and B, whose projections are a and b; $a\,b$ is the

line of centres, and C the point of contact, the line of contact being at right angles to the plane of projection, and passing through the point C. The two foci h and k revolve about a and b as centres, and for each complete revolution of the wheels, each describes a circle whose radius is equal to $a\,h$, which is twice the eccentricity of the ellipse.

Some of the properties of the ellipse have been already stated, and also various methods of drawing ellipses, which are shown on Plate II.

From the qualities stated in Art. 46, which we repeat here, it will be obvious that the following conditions exist. Referring to fig. 54, Plate II., in the ellipse ACBD, whose foci are F and H, we have, taking the point D and any other point, as O, on the circumference of the ellipse, $\mathrm{D\,F} + \mathrm{D\,H} = \mathrm{O\,F} + \mathrm{O\,H} = \mathrm{A\,F} + \mathrm{F\,H} + \mathrm{H\,B}\ (= \mathrm{H\,F} + 2\,\mathrm{A\,F}) = \mathrm{A\,B}$ the major axis.

In fig. 264 we have in the ellipse D, $\mathrm{C}\,h + \mathrm{C}\,a = p\,h + p\,a = l\,h + h\,a + a\,m\ (= a\,h + 2\,a\,m) = l\,m$ the major axis. From this it is clear that $a\,\mathrm{C} + \mathrm{C}\,b = l\,m$, fig. 264; also for any new position of the point of contact C, we have the sum of the radii $a\,\mathrm{C}$, $b\,\mathrm{C}$ a constant quantity and equal to the major axis $l\,m$. Therefore the wheels D and E will gear together in rolling contact.

In the position shown in fig. 264, the angular velocity-ratio, D being the driver, is $= \frac{b\,\mathrm{C}}{a\,\mathrm{C}}$, which is unity, because $a\,\mathrm{C} = b\,\mathrm{C}$. Suppose the wheel D to rotate in the direction indicated by the arrow, then the radius $a\,\mathrm{C}$ would increase until it became equal to $a\,l$, which is its greatest radius; at the same time, the radius $b\,\mathrm{C}$ would decrease until it became equal to $b\,l'$, which is its least radius. The velocity-ratio would now be represented by $\frac{bl'}{al}\left(=\frac{a\,m}{a\,l}\right)$ which is one limit.

For the next quarter of a revolution of the wheels the radius of D decreases, while that of E increases, until we have the velocity-ratio represented by unity, n and n' would then be in contact. In the next quarter of a revolution the radius of D further decreases until it is equal to $a\,m$, while that of E increases, and becomes equal to $b\,m'$; the velocity-ratio is now represented by $\frac{b\,m'}{a\,m}\left(=\frac{a\,l}{a\,m}\right)$.

The angular velocity-ratios for the four positions of the wheels when the points C; l and l'; n and n'; and m and m'; are in contact, are − unity, $\frac{a\,m}{a\,l}$, unity, and $\frac{a\,l}{a\,m}$.

Between each of these positions the velocity-ratio varies. Suppose the wheels D and E, fig. 264, have rotated in the direction indicated by the arrows from an initial position, in which $m\,m'$ coincided, the lines $l\,m$ and $l'm'$ would then be in one straight line, to the position shown in the figure, having C for the point of contact. During this motion the radius $a\,m$ has increased from $a\,m$ to a C, and the angle described by it is represented by $m\,a$ C, the radius $b\,m'$ has decreased to b C, and the angle described by it is represented by $m'b\,\mathrm{C} = m\,h\,\mathrm{C}$. The radius of the wheel D has therefore described an angle represented by a C h, greater than that described by the radius of the wheel E.

The angle $a\,\mathrm{C}\,h = m\,a\,\mathrm{C} - m\,h\,\mathrm{C}$; therefore the wheel D has "overtaken" the wheel E by the angle a C h, while E has "fallen behind" D by the same angle.

If we take another point of contact as p, and join $p\,a$, $p\,h$, then $a\,p\,h$ represents the angle by which the wheel D has overtaken the wheel E, supposing the motion of D to be that indicated by the arrow. The angle $a\,p\,h$ is less than a C h, which shows that the wheel D has now fallen behind E, and thus it continues to do until l and l' come into contact; when this takes place, each wheel has described two right angles, and during this time D has overtaken and fallen behind E by the angle a C h. And so on for each half revolution of the wheels, one of them alternately overtaking and falling behind the other.

269. Logarithmic Spiral.—Equal logarithmic spirals are rolling curves; they are, however, unsuitable for complete wheels, but portions, as $\frac{1}{4}$ or $\frac{1}{2}$ of a revolution of such spirals, may be employed when a corresponding amount of rotation is required, or a number of similar sections may be employed to build up a complete wheel.

A logarithmic spiral is a curve such that the tangent at any point makes a constant angle with the radius at that point; the radius being the line drawn from the point on the curve to the centre or pole of the curve. The curve

never reaches its pole, because this constant angle never vanishes, and hence this spiral is unsuitable for the pitch lines of wheels that are required to rotate continuously.

Two similar and equal portions of logarithmic spirals may be employed to transmit circular motion, the conditions that must exist, as stated in Art. 266, being fulfilled; that is, supposing the two spirals to be in gear, and D and E to rotate about their axes in rolling contact from the initial position shown in the figure, *a* C and *b* C being the radii in contact, until the radius *a p* is in the line of centres *a b*, with the radius *b q*; the arcs C *p* and C *q* must be equal; also *a p* and *b q* must coincide with the line of centres; *a p* and *b q* in fact forming the line *a b*. A portion of a logarithmic spiral may be employed to give motion to a sliding bar.

270. In fig. 265 is shown a pair of equal and similar logarithmic spirals D and E in gear; they are centered at their poles, the projections of which are marked *a* and *b*.

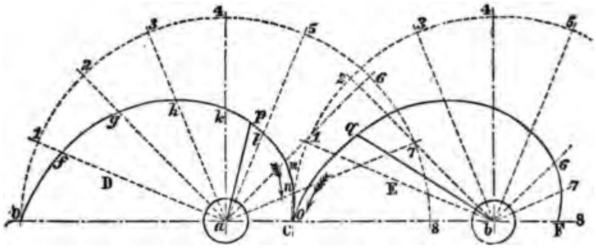


Fig. 265.

The point of contact C is in the line of centres *a b*. In the wheel D the semicircle 0...4...8, which contains the spiral, is divided into 8 equal parts, so that the angles 0 *a* 1, 1 *a* 2, 2 *a* 3, etc., are equal. Then by the property of the logarithmic spiral, the radii *a f*, *a g*, *a h*, etc., are in geometrical progression; that is,

$$\frac{a\,o}{a\,f}=\frac{a\,f}{a\,g}=\cdots\cdots\cdots=\frac{a\,n}{a\,\mathrm{C}},$$

Also

$$\frac{a\,o}{a\,k}=\frac{a\,k}{a\,\mathrm{C}}$$

$$a\,o \times a\,\mathrm{C} = (a\,k)^2$$

$$a\,k = \sqrt{a\,o \times a\,\mathrm{C}}.$$

Which gives an easy construction for finding points in the curve, as the following example will show:—Given the radii $a\ o$ and a C, required the logarithmic spiral $o \ldots$ C.

Find by the ordinary construction the mean proportion between $a\ o$ and a C, which is marked in the figure $a\ k$, $a\ k$ being at right angles to oaC; then find the mean proportion between $a\ o$ and $a\ k$, as $a\ g$, which bisects the angle $o\ a\ k$, and so on, any number of radii may be found and the curve drawn through the extremities of the radii.

The velocity-ratio varies from $\frac{\mathrm{C}\,b}{a\,\mathrm{C}}$ to $\frac{b\,\mathrm{F}}{a\ o}$, D being the driver, which is assumed as rotating in the direction indicated by the arrow.

271. Lobed Wheels.—In fig. 266 is shown a pair of lobed wheels D and E in gear, they are connected in their axes A and B, whose projections are a and b. The wheel D, which is the driver, is an ellipse, a being one of its foci; E is a two-lobed wheel, and b is its geometrical centre. The ellipse is called a one-lobed wheel when used as one of a set of lobed wheels, and it forms the foundation of the system, of which the rest, consisting of two or more lobes, is built up. The discovery of the properties of these wheels was made by the Rev. H. Holditch. Lobed wheels fulfil the conditions stated in Art. 266, page 223; thus, in fig. 266, we have the arc Cf = arc Cg, and radii $a\,f + b\,g = a\,b$, the line of centres.

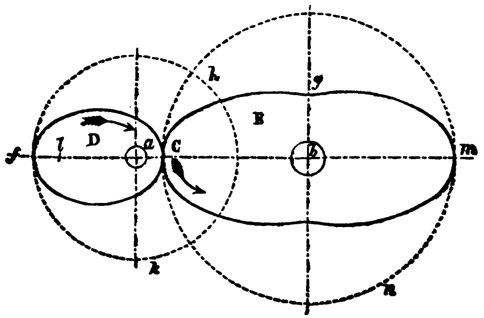


Fig. 266.

If D is the driver, and it rotates as indicated by the arrow,

the angular velocity-ratio $\frac{\mathrm{D}}{\mathrm{E}} = \frac{b\,\mathrm{C}}{a\,\mathrm{C}}$ for the position shown in the figure, and it varies between that and $\frac{b\,g}{a\,f}$ for each revolution of the driver. In lobed wheels, the number of lobes of the two wheels in gear indicate the number of revolutions respectively of the shafts or their velocity-ratio. Thus, in the present case, the velocity-ratio of the shafts A and B, A being the driver, is $\frac{2}{1} = 2$. If a pair of wheels of 4 and 5 lobes respectively were in gear, then we should have the velocity-ratio $\frac{\mathrm{A}}{\mathrm{B}} = \frac{5}{4}$, and so on, just the same as would be the case with a pair of wheels having the same number of teeth as there are lobes in the lobed wheels. The dotted circle $f\,h\,k$ is the path of the point f of the wheel D; and the circle C $m\,n$ that of the point C or m of the wheel E. The difference between the greatest and the least radii is called the inequality; in fig. 266 this difference is for the ellipse, the distance $a\,l$, which is the distance between its foci, and for the bi-lobe E, C b – $b\,g$, which is equal to $a\,l$. All lobed wheels of the same inequality will work together in rolling contact.

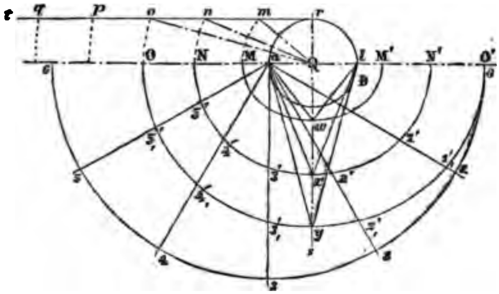


Fig. 267.

272. In fig. 267 is shown a construction for obtaining the necessary radii for wheels of any number of lobes. As before stated, the ellipse forms the foundation or base for these wheels.

Let the ellipse D, of which only one-half is shown, be one of a set of lobed wheels, of which we require, say, wheels of

2, 3, 4, and 5 lobes. Let *a* and *l* be the foci of the ellipse D, and Q its geometrical centre. From Q as a centre, describe a semicircle *a r l*, with the semifocal distance or half inequality Q *a* as a radius, and through Q draw *r* Q *s* at right angles to *a l*, cutting the semicircle *a r l* in *r*. From *r* draw a tangent to the circle parallel Q *a*. From Q as a centre with Q M, half the major axis of the ellipse D, as a radius, describe an arc of a circle M *m*, cutting the tangent *r t* in *m*, and from *m* set off along *r t*, *m n*, *n o*, etc., each equal to *r m*. From Q, with radii Q *n*, Q *o*, etc., describe arcs of circles cutting Q *o* in N 0, etc. Then Q *n*, Q 0, etc., are the semi-major axes of the ellipses, out of which are to be formed the 2, 3, etc., lobed wheels E, F, etc. Make Q M′ Q N′, Q O′, etc., equal to Q M, Q N, Q O, etc., and complete the concentric semi-ellipses, whose major axes are N N′, O O′, etc., and common foci *a*, *l*; see the ellipse in previous articles.

Describe a semicircle from *a* as a centre with a radius great enough to contain the largest semi-ellipse; in the figure we have shown the ellipses for the two and three lobed wheels, divide this semicircle into any convenient number of equal arcs, and draw radii; in the figure we have divided it into six parts, and numbered the radii 0...6. This number, however, must be increased when setting out the patterns, or even larger figures than those shown, so as to obtain more accurate curves. We will now proceed to draw a pair of wheels from fig. 267.

273. Suppose we require a pair of lobed wheels E and F, of two and three lobes respectively. Draw the centre line *a a*, fig. 268, and fix upon the point *b*, which is a projection of the axis B of the wheel E. From *b* as a centre, with a radius *a* N′, fig. 267, describe a circle N′D *v*, N′ and D being in the line of centres *b c*, and through *b* draw *u b v* at right angles to *a a*. Divide each of the quadrants, into which the lines N′ D, *u v* divide the circle, into six equal arcs, and draw radii 1 *b*, 2 *b*, etc. The radii *b* N′, *b* D are each equal to *a* N′ in fig. 267; make *b* I, *b* II,...*b* VI, in the radii *b* 1, *b* 2,...*b* 6 equal to the radii *a* 1′, *a* 2′...*a* N, fig. 267. Through these points draw the curve D VI N′ D, which is the trace of the pitch surface of the bi-lobe E; the construction is shown for one-quarter only, but the remaining quarters are exactly similar.

For the tri-lobe F mark off along the line of centres D c equal to a O, fig. 267, and so determine c, a projection of the axis C, upon which is centered the wheel F. Make c O′ equal to a O′, fig. 267, then c D and c O′ are the least and greatest radii of the wheel F. From c as a centre, with radius c O′,

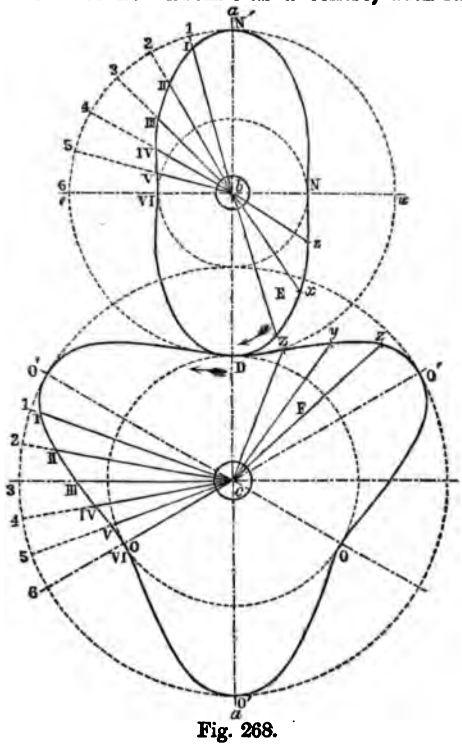


Fig. 268.

describe a circle and divide it into six equal arcs, and draw radii c O′, c O′, c O, c O, making the two former equal to O′, and the two latter to c D. Divide each of the six parts into which the circle is now divided into six equal arcs, and draw radii, as shown in the segment O′ 3 6. Make C I, C II,...C VI in the radii C 1, C 2,...C 6, equal to a 1′, a 2′,

...a O, fig. 267. Through the points O′...VI thus obtained draw the curve O′...VI, which is one-sixth of the trace of the pitch surface of the wheel F; the remaining five portions are exactly similar. If we draw lines from the foci a and l, fig. 267, to the points w, x, and y of the ellipse, these lines are the mean radii of those ellipses. The radii are shown in fig. 268, also the extreme and intermediate radii b Z, b Z of E and C Z, C Z with the points of contact.

Suppose E to be the driver, and to rotate in the direction indicated by the arrow, the angular velocity-ratio of $\frac{E}{F} = \frac{cD}{bD}$ which is its least value, D being the point of contact. The ratio increases until it reaches its greatest value, when O′ and N are in contact.

The construction of the 4, 5, 6, etc., lobed wheels will be obvious from these two examples. The necessary semi-ellipse, fig. 267, must first be drawn, and then a semicircle described with the greatest radius of the ellipse as a radius, this circle being divided into any convenient number of equal parts. Then a circle of the same radius as this circle must be described, and its circumference divided into twice as many arcs as there are to be lobes in the wheel; and each of these arcs is to be divided into the same number of equal parts as the semi-ellipse is divided into. The rest of the construction follows immediately from the examples given.

SECTION V.

WHEELS IN PAIRS AND IN TRAINS.

274. A train of wheel work is generally understood to mean a combination of wheels and axes in gear, and arranged for some specific object, as to vary the velocity-ratio between two shafts in such a way as could not be done conveniently by a simple pair of wheels, or wheel and pinion, or to vary the amount of power as it passes through the train. Of course, where one condition is fulfilled, the other is frequently

the only one, either speed or power, that has to be considered. The simplest case of a train of wheels is where there are but two wheels and two axes; usually such an arrangement is not termed a train; but, as similar calculations are involved, we may consider this as one limit of the system known as a train of wheel work, and consider it in the first instance. In all cases we shall assume the wheels as circular; that is, wheels having right circular cylinders for their pitch surfaces, the traces of which on a plane perpendicular to their axes are circles, called pitch circles, as spur wheels. If bevel wheels are needed, then no difficulty need present itself, as a reference to the articles on bevel wheels will show.

275. Wheels in Pairs.— Two wheels, or a wheel and pinion fixed on separate axes, constitute a pair of wheels; these wheels may be used to vary either the velocity-ratio of the connected axes, or the amount of power transmitted by them; we have just remarked that the two things go together, but are not always to be considered, as there are cases where one of them may be left out of consideration. A pair of wheels may also be employed simply to connect two shafts, but when one shaft is required to turn in the opposite direction to the other, it is usual to employ two equal and similar wheels.

276. In fig. 269 is shown two axes A and B connected by a pair of wheels; A is the driver, rotating in the direction indicated by the arrow. A *a* and B *b* are radii, which are equal to A C and B C respectively. The angular velocity-ratio $\frac{A}{B}=\frac{BC}{AC}$; in Art. 233, page 192, we have shown how this is connected with the number of teeth, etc. From what is stated in the articles referred to, certain problems that present themselves can easily be solved; however, we will take an example or two of common occurrence.

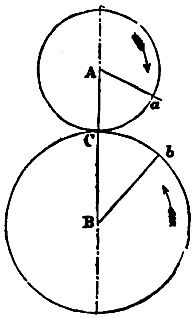


Fig. 269.

I. Suppose A, fig. 269, has 20 teeth, and makes 48 revolutions per minute; required the number of teeth in B for it to

make 15 revolutions in the same time. In Art. 233, we have

$$\frac{N}{n}=\frac{R}{r}=\frac{v}{V}=\frac{e}{E}\text{..................(1)}$$

from which we obtain

$$\frac{N}{n}=\frac{e}{E}\text{..........................(2)}$$

Substituting the given values of N, E, etc., and multiplying up, we have

$$n\,e=N\,E$$
$$e=\frac{20\times 48}{15}=64,$$

which is the number of teeth that B must have. The radii of the wheels, and hence the distance A B, depends upon the pitch, which, when determined, the other quantities are easily found by equation (3), Art. 233; see also Art. 234.

II. Suppose the velocity-ratio $\frac{A}{B}=3$, and B has 60 teeth; required the number of teeth in A. From equation (1) we obtain

$$\frac{V}{v}=\frac{n}{N}\text{.............................(3)}$$

Substituting the given values and transposing, we have

$$\frac{60}{N}=3$$
$$N=\frac{60}{3}=20.$$

III. Suppose the radius of A is 10 inches, that of B 55 inches, and A makes 40 revolutions per minute; required the number of revolutions of B in the same time.

From equation (1) we have

$$\frac{R}{r}=\frac{e}{E}\text{............................(4)}$$

Substituting the given values, we have

$$\frac{10}{55}=\frac{e}{40}$$
$$e=\frac{400}{55}=7\cdot 272$$

IV. Suppose A makes 22 revolutions per minute, and B 106; required the least number of teeth in the wheels.

From equation (1) we have

$$\frac{N}{n}=\frac{e}{E}\quad\text{..............................}\quad(5)$$

$$\frac{e}{E}=\frac{22}{106}$$

If the numbers 22 and 106 are prime to each other, then these numbers are the least numbers of teeth of the wheels; but if they are not, divide each by the greatest common measure or greatest common divisor of the numbers, and the quotients will be the required numbers.

The greatest common measure of two numbers is found as follows:—Let 22 and 106 be the numbers, divide the greater by the less until the remainder is less than the divisor, then divide the divisor by the remainder and obtain a quotient and a remainder; repeat the operation until a remainder is obtained that will divide the previous divisor and leave no remainder. The remainder thus obtained is the required G.C.M.; if this remainder is unity, the two numbers are said to be prime to each other.

Example.—

```
Divisor, 22)106(4, Quotient
             88
Remainder,   18)22(1, Quotient
                18
        R.,      4)18(4,   Q.
                   16
        R.,         2)4(2, Q.
                      4
```

Here 2 is the last remainder, and it is contained twice in the previous divisor, hence it is G.C.M. of the numbers 22 and 106. If we divide the numerator and denominator of the fraction $\frac{22}{106}$ by their G.C.M. we obtain $\frac{11}{53}$; therefore 11 and 53 are the least numbers of teeth that can be used for the wheels B and A respectively.

277. Hunting Cog.—In the example of the previous article, we have a pair of wheels A and B, of 53 and 11 teeth respectively; these numbers 53 and 11 are prime to each other, so that if two particular teeth T and *t*, one in each wheel, were in contact at a certain instant, these two teeth would not

come into contact again until A makes 11, or B makes 53 revolutions. From this it is easy to see that 53×11 pairs of teeth must pass a given point, as C, before T and t come into contact again.

If the numbers of teeth are not prime to each other, as 64 and 16, then divide each by the G.C.M. and so obtain two numbers that are; the numbers of revolutions and the numbers of pairs of teeth can then be found as before.

If instead of the numbers 53 and 11 we had, say, 64 and 16, then as the smaller is a divisor of the larger, the two teeth T and t will come into contact oftener than if the numbers 64 and 16 were prime to each other. This frequency of contact is considered to be injurious to the wear of the teeth of wheels employed for mill and machine purposes, and it is avoided by adding one tooth, called the hunting cog, to the larger one, so as to obtain a pair the number of whose teeth are prime to each other, as 65 and 16.

However, this addition of a tooth changes slightly the velocity-ratio, and in cases where that is inadmissible, as in the change wheels of a screw-cutting lathe, the hunting cog cannot be employed; but in other cases it may, and is to a considerable extent.

The idea of the use of the hunting cog is, that it enables the teeth to wear more evenly, as two particular teeth, if one is faulty, are brought into contact the least possible number of times in a pair of wheels that have to transmit a velocity-ratio approximating as near as possible to what that velocity-ratio would be if the hunting cog did not exist.

278. Wheels in Trains.—A train of wheel work is represented in fig. 270; there are four axes A_1, A_2, A_3, A_4. On the first axis A_1 there is a pinion, which is the driver of N_1 teeth, gearing into a wheel on the axis A_2 of n_1 teeth; on this axis is a pinion of N_2 teeth, gearing into a wheel on the axis A_3 of n_2 teeth; and on this shaft is a pinion of N_3 teeth, gearing into a wheel on the axis A_4 of n_3 teeth. We have thus four axes and three pairs of wheels and pinions; the first and last axis carrying only a pinion and a wheel respectively, while the second and third each carry a wheel and pinion. It is clear that the velocity-ratio of the axes A_1 and A_2 would be the same whether the axes A_3

and A_4 existed or not, supposing a given motion be given the driver A_1; and, further, it is clear that the velocity-ratio $\frac{A_2}{A_3}$ depends upon the numbers of the teeth N_2 and n_2 and the motion given to A_2; and, finally, the velocity-ratio $\frac{A_3}{A_4}$ depends upon N_3 and n_3 and the motion given to A_3.

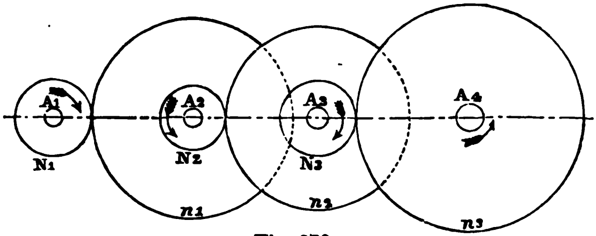


Fig. 270.

We have simply a combination of three pairs of wheels, and the velocity-ratio $\frac{A_1}{A_4}$ is found by multiplying together the teeth in the driver for a numerator and those in the followers for a denominator, thus—

$$\text{Velocity-ratio, } \frac{A_1}{A_4} = \frac{n_1 \times n_2 \times n_3}{N_1 \times N_2 \times N_3}$$

Example.—Let $N_1 = 20$, $N_2 = 15$, $N_3 = 18 : n_1 = 60$, $n_2 = 45$, $n_3 = 72$, then

$$\text{Velocity-ratio, } \frac{A_1}{A_4} = \frac{\overset{3}{\cancel{60}} \times \overset{3}{\cancel{45}} \times \overset{4}{\cancel{72}}}{\cancel{20} \times \cancel{15} \times \cancel{18}} = \frac{36}{1} = 36;$$

that is, A_1 makes 36 revolutions to 1 of A_4. It is obvious that if we take the pairs separately we shall get the same result. Thus—

$$\text{Velocity-ratio, } \frac{A_1}{A_2} = \frac{n_1}{N_1} = \frac{60}{20} = 3.$$

$$\frac{A_2}{A_3} = \frac{n_2}{N_2} = \frac{45}{15} = 3.$$

$$\frac{A_3}{A_4} = \frac{n_3}{N_3} = \frac{72}{18} = 4.$$

$$\frac{A_1}{A_4} = 3 \times 3 \times 4 = 36.$$

It is also clear that the order in which the pairs of wheels occur in the train does not effect the result; thus we may interchange, supposing the pitches of the teeth were the same, the axes A_2 and A_3, and still maintain the same velocity-ratio for $\frac{A_1}{A_4}$.

Generally—Let there be m pairs of wheels, with teeth $N_1, N_2, \ldots N_m$, and $n_1, n_2 \ldots n_m$; and therefore there are $m+1$ axes, $A_1, A_2, \ldots A_{m+1}$, as there is always one axis more than there are pairs of wheels. Thus we have the velocity-ratio

$$\frac{A_1}{A_{m+1}}=\frac{n_1\times n_2\times \ldots n_m}{N_1\times N_2\times \ldots N_m}.$$

Example.—Let $m=5$; $N_1=10$, $N_2=12$, $N_3=15$, $N_4=12$, $N_5=10$; $n_1=24$, $n_2=30$, $n_3=40$, $n_4=36$, $n=25$. Then velocity-ratio,

$$\frac{A_1}{A_5}=\frac{\overset{2}{\cancel{24}}\times\overset{2}{\cancel{30}}\times\overset{4}{\cancel{40}}\times\overset{3}{\cancel{36}}\times 25}{\cancel{10}\times\cancel{12}\times\cancel{15}\times\cancel{12}\times 10}=\frac{1200}{10}=120.$$

There are six axes and five pairs of wheels, and the sixth axis makes 120 revolutions to one of the first. If the number of the axes be odd, then the last one turns round in the same direction as the first; and if the number be even, the first and last turn in opposite directions.

CHAPTER VII.

ON THE TEETH OF WHEELS.

Spur and Bevel Wheels: the Forms of their Teeth—Examples in Drawing—The Odontograph.

279. It is now proposed to consider the teeth of wheels, the various forms given to them, the methods of drawing them, and their strength.

We have already stated the object for which teeth are employed, and also pointed out that the object to be kept in view, when designing teeth, is to maintain as nearly as possible the perfect rolling of the pitch surfaces. Of course these pitch surfaces have no real existence, but it is convenient to assume that they have; and upon this assumption all our calculations respecting the transmission of motion with a certain fixed velocity-ratio will depend. As this perfect *rolling motion* of the ideal pitch surfaces of a pair of wheels in gear has to be produced by the *sliding motion* of the teeth, it will at once be seen that their form is of considerable importance, for upon the correctness depends how near we approach to this true rolling motion.

280. The form of the teeth depends upon the kind of motion communicated by the pieces in gear, as for instance for two rotating wheels, or for a rotating wheel and sliding rack, and also upon the kind of curve employed to describe the teeth. We shall consider the cases mentioned, and take such examples as will explain the kinds of curves given to the teeth for ordinary purposes, as the limits of this work will not permit of an investigation into all the cases.

The curves which form the acting surfaces of the teeth of wheels are the *cycloidal curves*, and the *involutes of circles.* In previous articles we have defined these curves, and have shown how each can be drawn. Let the student again refer to Plate III. for a few moments and the remarks thereon.

SECTION I.

CYCLOIDAL FORMS OF TEETH.

281. There are three cases of the cycloidal form of teeth; we shall consider the two commonest. The first is that in which there are two describing circles for every pair of wheels which are to gear together; and the second is that of a constant describing circle for all the wheels of the same pitch.

282. CASE I.—Let A and B, fig. 271, be the centres of a pair of wheels in gear, p C, p C their pitch circles, A B the line of centres, C the point of contact of the pitch circles, and A a C, B b C the describing circles. Let the circle A a C describe the epicycloid C c, and the hypocycloid C d; and let the circle B b C describe the epicycloid C e, and the hypocycloid C f. The diameters of the circles A a C, B b C being equal to the radii of the pitch circles of A and B respectively, the hypocycloids are straight lines, and are portions of the diameters of AaC and BbC. Now it can be proved that if the teeth have these curves for their acting surfaces, or surfaces of contact, and one wheel is made to rotate and thus drive the other by the sliding contact of their teeth, the angular velocity-ratio of the axes A and B is the same as if the two pitch circles rolled together as smooth wheels without sliding.

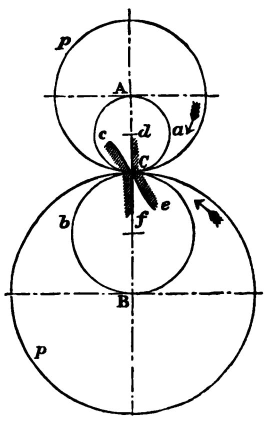


Fig. 271.

If R and r be the respective radii of A and B, then we have the velocity-ratio $\frac{V}{v} = \frac{r}{R}$.

Let R = 2, r = 3, then the velocity-ratio = $\frac{3}{2}$, and what we state is, that this velocity-ratio of the axes A and B is maintained by the sliding action of the teeth, which are shaped as described. We shall consider A to be the driver, but

B may be the driver if the teeth are formed in the manner described, except in certain extreme cases of velocity-ratio.

283. If the wheels rotate as shown by the arrows, and A is the driver, the contact between each pair of teeth takes place along the arcs *a* C, C *b*; the contact that takes place along *a* C *b*, before the teeth pass the line of centres A B, is called *approaching contact*, and that which takes place after passing A B, *receding contact*. The extent of action of a pair of teeth both before and after they pass the line of centres, depends upon the lengths of their faces, which, as we have already stated, are fixed for each pitch; but these may be varied, if circumstances require, so as to give any desired extent of path of contact. The flanks being made sufficiently long to allow the faces to work with freedom.

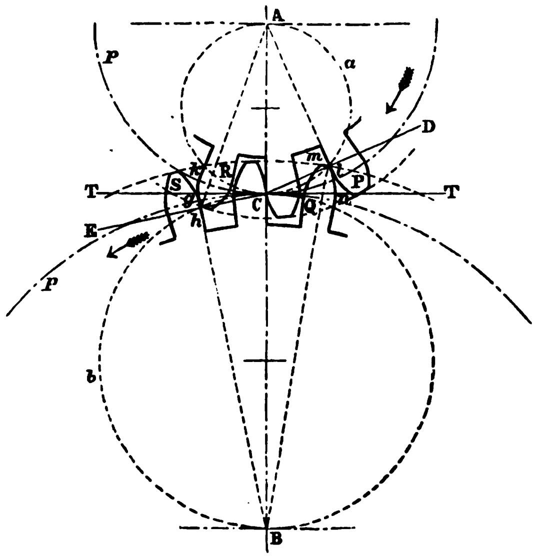


Fig. 272.

284. The *path or locus of contact* is the line along which contact between a pair of teeth takes place. In fig. 272 this line consists of the arcs *m* C, C *h* of the describing circles.

If A is the driver, and the wheels rotate as shown by the arrows, contact begins at *m* on the semicircle A *a* C, and continues along that arc to C; at the instant the teeth cross the line of centres A B, the direction of action is along the common tangent T T, which is at right angles to A B; the direction of the path then changes into the arc C *h* of the semicircle C *b* B, and finally ends at *h*.

Through *m* and *h* draw the lines C D, C E respectively, normals to the surfaces of the teeth in contact at *m* and *h*; then the angle D C T is the greatest *angle of obliquity* during approaching contact, and E C T that during receding contact. In the figure these angles are unequal, the reason for which will be given presently.

285. The *arc of contact* is the arc of the pitch circle which passes the point C in the line of centres A B, fig. 272, during the contact of a pair of teeth. This arc is divided into approaching and receding contact. In fig. 272, *l* C is the arc of approaching contact, and C *h* that of receding contact, assuming the wheel A to be the driver, and to rotate as shown by the arrow. To transmit the motion of A to B, it is necessary that at least one pair of teeth shall always be in contact, and, therefore, the length of the arc of contact must be greater than the pitch; and if practicable, it should be long enough to allow of two pairs of teeth to be always in contact.

The action that takes place between a pair of teeth after passing the line of centres, is considered to be of a less injurious character than that which takes place before passing that line.

In fig. 272 is shown a portion of a pair of spur wheels in gear, the teeth of which are formed as just described. Let A, as before, be the driver, and let the arcs of approaching and receding contact each equal the pitch. Make the arc C *l* on the pitch circle *p* C of A equal the pitch; through *l* draw the hypocycloid *l* A, meeting the semicircle A *a* C in *m*; and through *m* draw the epicycloid *m n*. Then *m* is the point in which the two teeth P and Q come into contact; *m n* is the length of the face of the follower's tooth; and *m l* that of the driver's flank, which are used during approaching contact.

Make the arc C *g* on the pitch circle of *p* C of B equal the pitch; through *g* draw the hypocycloid *g* B, meeting the

semicircle C b B in h; and through g draw the epicycloid $g\,k$. Then h is the point in which the two teeth R and S cease contact; $h\,k$ is the length of the face of the driver's tooth; and $g\,h$ that of the follower's flank which are used during receding action. A m, A h are the respective lengths of the radii of the driver at beginning and end of contact, and B m, B h those of the follower.

It will be seen that only portions of the flanks are used, while the whole of the faces come into contact, and these portions of the flanks are much smaller than the faces. And further, the length of the face of the follower is greater than that of the driver for equal arcs of approach and recess, and this is because the driver is the smaller wheel of the two, to which point we now refer.

Let B be the driver, and let the arcs of contact be the same as before; then g C is the arc of approach, and C l that of recess, assuming the wheels to rotate in the directions opposite to those shown by the arrows. Through m and h draw the normals C D, C E, and through C draw the common tangent T T. Now it can be proved that the arcs C l, C m, C n, C g, C h, C k are all equal; and therefore, as the circle B b C is greater than the circle A a C, it follows the angle D C T is greater than the angle E C T; therefore the face of the tooth of B is longer than that of A.

If therefore the faces of the teeth of the driver and follower are made equal in length, which is usually the case, it follows that, for a given pair of wheels, if the smaller is the driver, the arc of receding contact is greater than if the larger is the driver; and hence a smaller wheel can be employed to drive than to follow.

Professor Willis has calculated and arranged in a table the limiting numbers of teeth which can be employed in a pair of wheels having a given arc of receding action, according as the smaller or the larger is the driver. From this table we give the following example referring to spur wheels:—Let the arc of receding action equal the pitch, and let there be 12 teeth in the pinion; if the *wheel* drives, then 30 is the least number of teeth it can have, but if the *pinion* drives, a wheel of 19 teeth may be employed. It must, however, be noticed, that these values assume the tooth and space each equal half the

pitch, so that in practice a greater number of teeth must be taken in every case.

286. The curves for the teeth of internal or annular wheels are given in Table XV.; the setting out of those curves is similar to the case of spur wheels already considered. We therefore refer the student to that case and the examples that follow. In the example named, the teeth are formed as in Case II., but it will be easy to apply the example to Case I.

The curves for the teeth of racks and pinions are also given in the table, and in a succeeding paragraph an example is worked out.

287. The following table gives a summary of the kinds of curves used for Cases I. and II.

TABLE XV.

CURVES FOR THE TEETH OF WHEELS.

I. Diameter of describing circle equal to the radius of the pitch circle; for a pair of wheels there are two describing circles, which are equal to the radii of the pitch circles.

Kind.	Top of Tooth	Bottom of Tooth.
a. External wheels,	Epicycloid,	Hypocycloid.
b. Pinion and	Epicycloid,	Hypocycloid.
Internal wheel,	Hypocycloid,	Epicycloid.
c. Rack and	Cycloid, diameter of, describirg circle equal to radius of pitch circle of pinion,	Straight line perpendicular to pitch line.
Pinion,	Involute of pitch circle,	Hypocycloid.

II. Describing circle constant for each pitch, and equal to the radius of the pitch circle of a pinion of 12 teeth.

Kind.	Top of Tooth.	Bottom of Tooth.
a. External wheels,	Epicycloid,	Hypocycloid.
b. Pinion and	Epicycloid,	Hypocycloid.
Internal wheel,	Hypocycloid,	Epicycloid.
c. Rack and	Cycloid,	Cycloid.
Pinion,	Epicycloid,	Hypocycloid.

288. Case II.—Let A and B, fig. 273, be the centres of a pair of wheels in gear, *p* C, *p* C their pitch circles, A B the line of centres, C the point of contact of the pitch circles, and D *a* C, E *b* C the describing circles, which are equal in diameter. Let the circle D *a* C describe the epicycloid C *c*, and the hypocycloid C *d*; and let the circle E *b* C describe the epicycloid C *e* and the hypocycloid C *f*.

Then it can be proved that these curves will slide together and maintain the same angular velocity-ratio of the axes A and B, as if the two pitch circles rolled together. Therefore if the teeth are formed with such curves for their acting surfaces, the constant angular velocity-ratio of the axes A and B can be maintained by the sliding motion of those teeth. And further, if in place of A or B we substitute a wheel of a different diameter to either A or B, and form the teeth of such other wheel by the constant describing circle D *a* C or E *b* C, then the two wheels in contact will work correctly together, and maintain the same angular velocity-ratio as if their pitch circles rolled together. The only limit to the correct working of such wheels is that they must be all of the same pitch. In practice it is found that the diameter of the rolling circle should not be greater than the radius of the pitch circle of a pinion of 12 teeth, and in that case such a pinion would have radial flanks; and it must not be greater than the radius of the smallest wheel in the set.

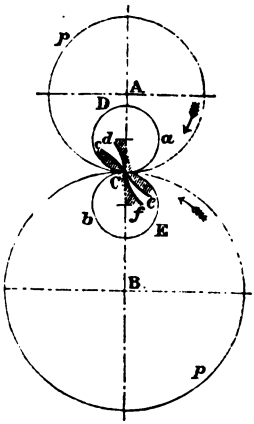


Fig. 273.

The rule is to make the diameter of the constant rolling circle equal to the radius of the pitch circle of a pinion of 12 teeth; and of course the diameter of the rolling circle varies for each pitch.

The credit of having first shown this property of a constant describing circle for all wheels of the same pitch belongs to

Professor Willis. As in Case I., either A or B may drive, but if the smaller drives and the arcs of approach and recess are to be equal, then the faces of the teeth of the smaller must be longer than those of the follower; we will take an example which will show this.

289. In fig. 274 is shown a portion of a pair of spur wheels in gear, the teeth of which are described as stated in the previous article.

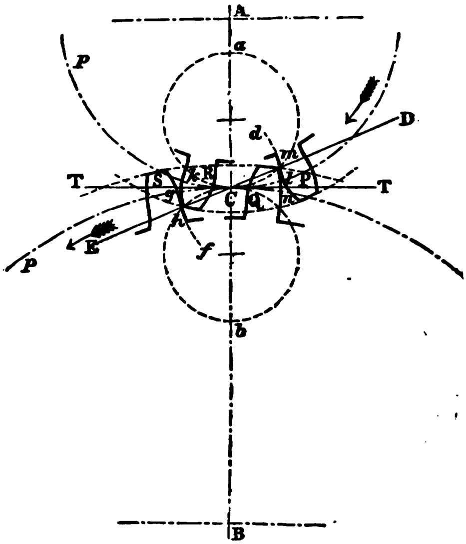


Fig. 274.

Let A be the driver, which rotates as shown by the arrow, and let the arcs of approaching and receding contact each equal $\frac{3}{4}$ the pitch. Draw the describing circles *a m* C, *b h* C, each equal in diameter to the radius of the pitch circle of a pinion of 12 teeth of the same pitch as A and B. Make the arc C *l* on A's pitch circle *p* C equal $\frac{3}{4}$ the pitch; through *l* draw the hypocycloid *l d*, meeting the circle *a m* C in *m*; and through *m* draw the epicycloid *m n*. Then *m* is the point in which the two teeth P and Q come into contact; *l m* is the

length of the flank of the driver; and $m\,n$ that of the face of the follower which are used during approaching contact.

Make the arc C g on B's pitch circle p C equal $\frac{3}{4}$ the pitch; through g draw the hypocycloid $g\,f$, meeting the circle $b\,h$ C in h; and through h draw the epicycloid $h\,k$. Then h is the point in which the teeth R and S cease contact; $h\,k$ is the length of the face of the driver; and $g\,h$ that of the flank of the follower which are used during receding contact.

If B is the driver, and if it rotates in the direction opposite to that indicated by the arrow, then g C is the arc of approach, and C l that of recess (see fig. 275). It can be proved that the arcs C g, C h, C k, C l, C m, C n, are all equal; and as the circles $a\,m$ C, $b\,h$ C are equal in diameter, the line D E drawn through the points of contact $h\,m$ of the teeth, is a common normal to the surfaces of the teeth at h and m. Therefore the perpendiculars from h and m upon the common tangent to the pitch circles at C are equal; but as the pitch circle of A is smaller in diameter than that of B, the face $h\,k$ of the tooth R is longer than the face $m\,n$ of the tooth Q. It therefore follows that for equal arcs of recess, the faces of the teeth of the smaller

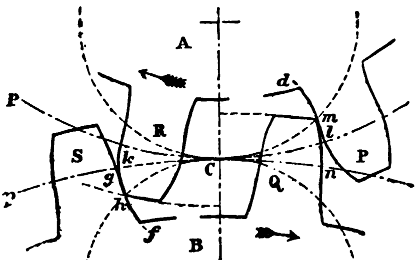


Fig. 275.

wheel must be longer than those of the larger. Hence, if the faces of the teeth of the driver and follower are of equal length, which is usually the case, a larger wheel may be employed to drive than to follow. This is the reverse of what we have stated for Case I., where a smaller wheel may drive than follow.

290. In fig. 275 are shown the spur wheels of fig. 274, drawn to double the scale; the usual proportions of the teeth are employed, which make a difference in the lengths of the arcs of contact, that on the right of the centre line being the greater.

If the smaller wheel A is the driver, and if it rotates in the direction shown by the arrow, C *g* is the arc of receding contact, and *l* C that of approaching contact; the arc *l* C is greater than the arc C *g*. If the larger wheel B is the driver, and if it rotates in the direction shown by the arrow, then *l* C is the arc of recess and *g* C that of approach; therefore, if the larger wheel drives, the arc of recess is greater than when the smaller is the driver.

291. The curves for the teeth of internal or annular wheels, and of racks and pinions, are given in Table XV., page 244. In Art. 308, page 263, is given an example of a pinion and an internal wheel in gear. The teeth of bevel wheels may be formed by either Case I. or II.; in Art. 310, page 264, is given an example of a pair of bevel wheels in gear, the teeth of which are described according to Case I.

292. The following properties are possessed by wheels whose teeth are described according to Cases I. and II. :—

Case I.—If in a pair of wheels which have the faces of their teeth of the same size, the smaller is the driver, then the arc of receding contact is greater than in Case II.; but *vice versâ* if the larger is the driver. If the patterns of the wheels have to be made by hand, then there is less work in forming the teeth; but as this rarely happens at the present time, owing to the introduction of wheel-cutting engines and of wheel-moulding machines, we may almost pass it over. The great disadvantages are, that no two wheels of the same pitch will work together correctly, except they are pairs; that is to say, suppose a pair of wheels of 20 and 50 teeth, and a pair of 30 and 70 teeth are made, then the 20 and 50 and the 30 and 70 will work together correctly, but not the 30 and 50 or the 20 and 70; this is a very important matter, as a question of economy in pattern-making. Again, the flanks of the teeth being radii of the pitch circle, the portions of the teeth inside the pitch circle are narrower than those on the pitch circle; the teeth are therefore not so strong as those of Case II.

If in the case of racks and pinions the pinion is the driver, the arc of recess of the rack is a straight line, and therefore the action is confined to the point of the tooth represented by the pitch line, as C, fig. 2, Plate XII., instead of being distributed over a portion of the flank.

CASE II.—Any two wheels of the same pitch will work together correctly, which is a very important matter not only as regards pattern-making but also in construction, as a wheel may have to gear with two others of different diameters. The teeth are wider inside the pitch circle than those formed by Case I., and are therefore stronger.

As the flanks of the teeth of racks are cycloids, the receding contact, if the pinion is the driver, is not confined to the point of the tooth in the pitch line, but is distributed over a portion of the flank.

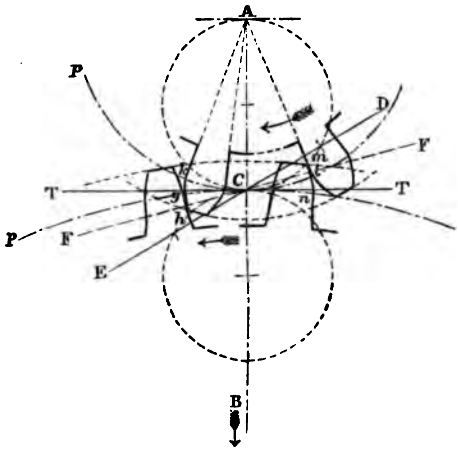


Fig. 276.

293. In fig. 276 is shown a pinion of 12 teeth in gear, with a larger wheel; the diameter of the describing circle is therefore equal to the radius of the pitch circle of the pinion. If the arcs of approach and recess are each equal to the pitch, the greatest angle of obliquity of approach or recess

is 30°; the line F F bisects the angle D C T; and the angle F C T, which contains 15°, is therefore the mean angle of obliquity. Professor Willis states, as the results of practical experience, that the mean angle of obliquity should not exceed 15°; a good value is between 14° and 15°; therefore, if the arcs of approach and recess are each equal to the pitch, the diameter of the describing circle must not be greater than the radius of the pitch circle of a pinion of 12 teeth, as shown in fig. 276; this pinion should be the smallest wheel in the set.

If we give the usual length of face to the teeth in the case considered, the arcs of approach and recess are less than the pitch, and therefore the greatest angles of obliquity are less than 30°. In fig. 276 the teeth are shown according to the proportions already given. In the following article we give the values of the greatest angles of obliquity of action in certain extreme cases.

294. In practice it is usual to use a constant length of face for the teeth of wheels; it is therefore important to notice the value of the greatest angle of obliquity of action in each of the extreme cases which occur. With this object we have arranged the following table, which gives the values of the greatest angles of approach and recess for the two cases of cycloidal teeth considered. The extreme cases which occur are, for ordinary external spur gearing, when two equal wheels are in gear, the wheels being the smallest in the set, say of 12 teeth; and when a pinion, say of 12 teeth, and a rack are in gear. All other cases will lie between these.

The constant length of face varies with different makers, as we have already stated; the most commonly employed are the following, where p stands for the pitch:—

$$p \times \cdot 33,\ p \times \cdot 35, \text{ and } p \times \frac{5\frac{1}{2}}{15} = p \times \cdot 3\dot{6}.$$

It will be obvious that the angle of obliquity of action depends upon the length of face employed; in the present case we have taken it equal to $p \times \cdot 33$.

It will be seen that the following table agrees with the statement previously made, that for Case I. a smaller wheel may be employed to drive than to follow; for in the extreme

case of a pair of wheels, where the pitch circle of the larger becomes a straight line, as in a rack, the greatest angle of obliquity of recess is nil; in fact, in the case of a pinion of 12 teeth, 3 inches pitch, driving a rack, the path of recess is a little over $3\frac{1}{2}$ inches; whereas, if the rack drives, the path of recess is less than $2\frac{1}{2}$ inches. If in Case II., taking the same rack and pinion, the pinion drives, the path of recess is nearly $2\frac{1}{8}$ inches; while, if the rack drives, it is nearly $2\frac{1}{2}$ inches; all of which goes to prove that a larger wheel may be employed to drive than to follow.

TABLE XVI.

ANGLES OF OBLIQUITY OF ACTION OF TEETH.

Cycloidal.	Greatest Angle of Approach.	Angle at line of Centres.	Greatest Angle of Recess.	Length of Path of Recess for 3″ Pitch.
CASE I.				
Equal Wheels of 12 Teeth,	21°	0°	21°	2·16″
Rack and Pinion, Pinion Drives,	25°	0°	0°	3·5″
Rack and Pinion, Rack Drives,	0°	0°	25°	2·5″
CASE II.				
Equal Wheels of 12 Teeth,	21°	0°	21°	2·16″
Rack and Pinion, Pinion Drives,	25°	0°	21°	2·16″
Rack and Pinion, Rack Drives,	21°	0°	25°	2·5″
Involute teeth have a *constant* angle of obliquity of action.				

SECTION II.

INVOLUTE FORM OF TEETH.

295. The cycloidal form of tooth is the one generally employed, but in some cases the involute form may be employed with advantage, as it possesses properties which are possessed by neither of the cycloidal forms considered. We will therefore state briefly what these properties are.

We have previously defined the involute of a circle (see Plate III. and text), and have shown how to draw it.

296. Let A and B, fig. 277, be the centres of a pair of wheels in gear, p C, p C their pitch circles, A B the line of centres, and C in A B the point of contact of the pitch circles. Through C draw D E inclined at any angle to A B; from A and B draw perpendiculars, A D, B E, upon D E; from A and B as centres, with radius A D, B E respectively, describe circles D H R, E L S; the line D E is a common tangent to these circles. Draw the involutes H C K, L C M of the circles D H R, E L S, and let them be in contact at C.

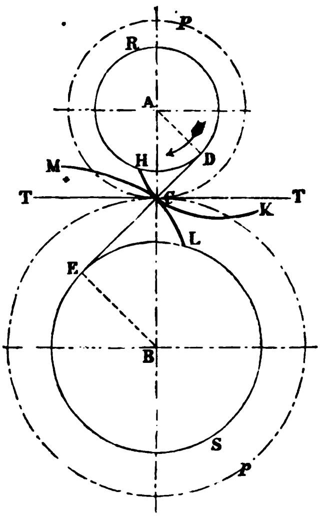


Fig. 277.

Let the wheel whose centre is A rotate in the direction shown by the arrow, and drive the wheel B by the sliding contact of the involutes H C K, L C M. Then it can be proved that the wheels will rotate with the same angular velocity-ratio as if the pitch circles rolled together.

The line D E, which is a common normal at C to the involutes, is the path or locus of contact of these involutes; that is to say, every point of contact of the involutes lies in that line.

The angle this line makes with T T, which is at right angles to A B, is the angle of obliquity. The inclination of the line D E may be varied at pleasure, and thus we may vary the diameters of the *base circles* D H R, E L S; as such alterations of D E will not alter the velocity-ratio, for the two triangles A D C, B E C are similar; therefore A D : B E : : A C : B C, also C D : C E : : A C : B C. But, as before stated, the angle D C T should be between 14° and 15°. This constant angle of obliquity of action is considered to be

injurious on account of the extra strain thrown upon the bearings of the wheels, and also on account of the increase of friction between the teeth in contact. In the cases of cycloidal teeth previously considered, we have seen that the angle of obliquity is greatest at the commencement and the end of contact, and that at the instant the teeth pass the line of centres it is nought; therefore the cycloidal form of tooth has this advantage over the involute form. But the involute form possesses the important property that the distance of the centres of a pair of wheels may be varied without destroying their correct working or their angular velocity-ratio; the only limit to this variation of centres being that the wheels remain in gear.

Also, as pointed out by Professor Willis, the amount of *back-lash* between the teeth may be varied at will. These two properties are important ones, and are not possessed by teeth of any other form.

All involute teeth of the same pitch work correctly together; they therefore possess the same property as those of Case II. of the cycloidal form.

It will be seen that the acting surfaces of involute teeth consist of single curves, which spread out at the root; whereas, in the cycloidal teeth, there are two curves, one for the face and one for the flank; the involute form is therefore a very strong form.

297. In fig. 1, Plate XI., is shown parts of a pair of spur wheels in gear, whose teeth are involutes of circles, as described in the previous article. The wheel, whose centre is A, is the driver; there are 25 teeth in A and 38 in B; the pitch measured on the arc of the pitch circle is $1\frac{1}{2}$ in. The angle of obliquity F C T is 15°. The usual proportions of the faces and flanks of the teeth are employed. Later on we shall give a general construction for determining these for a given length of contact.

Having calculated the diameters of the pitch circles by the usual formula,

$$D = \frac{P}{\pi} \times N,$$

draw the line of centres (Plate XI.) A B, the pitch circles

p C, p C, the line T T at right angles to A B, and the line F F inclined to T T at 15°.

From A and B let fall perpendiculars A D, B E upon the line F F; from A and B as centres, with radius A D, B E respectively, describe the base circles H D R, E L S; these base circles are the *evolutes* of the *involutes* to be drawn. Draw circles for the faces and flanks of the teeth; let the circle for the faces of A cut the line F F in h, and that for the faces of B cut F F in c. Then $c\,h$ is the path or locus of contact, which, in the example, gives an arc of contact a little greater than twice the pitch, so that the two pairs of teeth are always in contact.

In order that two pairs of teeth may always be in contact, the path or locus of contact, which is the common normal to the surfaces of the teeth at the point of contact, must not be less than twice the normal pitch. The normal pitch is the pitch as measured along the path of contact, and is found as follows:—In fig. 3, the lines A B, T T, etc., are situated similarly to the corresponding lines in fig. 3. Make C a in T T equal to the circular pitch, that is, the pitch as measured along the arc of the pitch circle, in the example it is $1\frac{1}{2}$ inches; from a draw $a\,b$ at right angles to and meeting C F in b; then C b is the normal pitch. As the triangle $a\,b$ C is similar to the triangle C D A, fig. 1, we have C b : C a : : A D : A C; that is, the normal pitch : the circular pitch : : radius of base circle : radius of pitch circle.

From any convenient points H, L, fig. 1, on the base circles, describe the involutes H K, L M. A portion of the involute H K is to be used for the acting surfaces of the teeth of A, and a portion of the involute L M for those of B. From C mark off, along the pitch circles, the pitch C l, C g, C k, C n, etc. Having determined the length of the path of contact $c\,h$, fig. 1, in the example it is determined by giving the usual length to the faces of the teeth, describe circles, H c R, O h N; from A and B as centres, with radius A c, B h respectively (A c nearly coincides with A D, and therefore the circle H D R with the circle H c R; they are shown as one circle in the figure). Then the circle H c R, O h N will cut the flanks of the teeth in the points where contact begins and ends, or ends and begins, according as one or the other wheel

drives. Through the points C, l, g, k, n, etc., draw the involutes for the teeth, meeting the circles G h P, Q c U in h, m respectively.

If A is the driver, and if it rotates as shown by the arrow, then c is the point where contact begins, and h is the point where it ends. The points c and m are upon the same circle, therefore $m\,n$ is the length of the face of B, and $m\,l$ that of the flank of A, which are used during approaching contact. The point h coincides with the point of intersection of G h P and F F, and is the point where contact ends; therefore $h\,k$ is the length of the face of A, and $g\,h$ that of the flank of B, which are used during receding contact. The portions of the flanks inside the circles H m R, O h N are straight lines, tangents to the involutes at m and h. As m is nearly on the base circle H D R, the portions $m\,d$ of the flanks of A may be considered as radial lines (A and B are here one tooth of wheel A and B).

298. To draw the portions $h\,f$ of the flanks of B, draw $h\,f$ parallel to E B, then $h\,f$ is the required line, which is a tangent to the involute at h. In the right-hand portion of fig. 1 is shown a construction for drawing the required tangents. From B as a centre with a radius B h, cut the involute L M in h'; make h' E′ equal to h E, E′ being on the base circle; join B E′. Then h' E′ B is a right angle, and h' E′ is a tangent to the base circle; draw $h'f'$ parallel to E′ B, then $h'f'$ is a tangent at h' to the involute, h' E′ being the normal. As a number of such tangents are required, describe a circle from B as a centre with radius Bf'; then lines drawn from the points of each side of the teeth in the circle O h N to touch this circle will be the required tangents, as $h\,f$. It must be observed that the tangents drawn for each tooth do not cross each other.

299. *General Case.*—Given the pitch circles, the angle of obliquity, and the length of the path of contact of a pair of spur wheels, to draw the teeth. Let the pitch circles be those of a pair of wheels of 40 and 30 teeth, of 1 inch pitch, with the angle of obliquity 15°, and having the length of path of contact equal twice the normal pitch, half on each side of the line of centres.

Fig. 2, Plate XI.—Draw the line of centres A B, the

pitch circle p C, p C (the centre of B is not shown, as it falls outside the sheet), the line T T at right angles to A B, and the line F F inclined to T T at 15°. Let A be the driver, and let it rotate in the direction shown by the arrow.

From A and B draw perpendiculars, A D, B E, upon the line F F; from A and B as centres, with radius A D, BE respectively, describe the base circles H D R, E S L. Mark off from C along D E, C h and C m each equal to the normal pitch, see fig. 3, where C b is the normal pitch, C a the circular pitch (= 1 inch in the present case), and where a b is at right angles to C b.

From A as a centre, with radius A h, describe the circle G h P; and from B describe the circle Q m U, with radius Bm; then these circles contain the extremities of the faces of the wheels A and B respectively. Through h and m draw the involutes $h\,g\,o$, $m\,l\,q$ meeting the circles G h P, Q m U in q, o. Then $h\,g\,o$ and $m\,l\,q$ are the portions of the involutes used for the teeth of the follower and driver respectively; the portions outside the pitch circle are the faces, and those inside are portions of the flanks.

From A and B as centres, with radius A m, B h respectively, describe the circles M m r, h e N; these circles will cut the flanks in the points where contact begins and ends. The remaining portions of the flanks are tangents to the involutes at the points where the circles M m r and h e N cut the curves, as at m and h; $m\,d$ and $h\,f$ being portions of these tangents. The tangents are drawn as described in Art. 298, p. 255. The extreme radii of the wheel A are A m at commencement, and A h at end of contact; those of the wheel B are B m and B h respectively.

The teeth of each wheel are usually made of an equal thickness on the pitch circle, but this thickness may be varied in extreme cases so as to give teeth of equal strength. It will be obvious that if the teeth are of equal thickness on the pitch circles, those of the smaller wheel of a pair will be narrower at the root than those of the larger. We have already shown that the portions $m\,d$ of the flanks are radial lines in the case of the smaller wheel of the set.

300. The normal pitch C m : the circular pitch C n :: AD : AC, therefore as the triangles A D C, fig. 1 and a b C,

fig. 3, are similar, and Ca, Cb equal the circular and normal pitches respectively, we have

$$Cb = Ca \times \text{cosine } \theta$$

(where θ denotes the angle C A D = angle a C b.)

Let $\theta = 15°$, and $Ca = 1$ inch, then

$$Cb = 1 \times \cdot 966 \text{ nearly} \qquad (\text{cos. } 15° = \cdot 96593.)$$

If m coincides with D, then the base circle coincides with the circle M m r, and the radius A$'$ m, which is drawn parallel to A D, is the least radius of base circle which can be employed for a given length of approaching contact, C m, A$'$ m being the radius of the pitch circle.

This determines the least number of teeth which can be employed in any wheel for a given half path of contact C m.

Let ϕ denote the angle A C D (= 90° − angle of obliquity), N the number of teeth in the smallest wheel of a set, and n the half path of contact C m, n being expressed in terms of the normal pitch as a unit. Then

$$N = 2\pi \times \text{tangent } \phi \times n \ldots\ldots\ldots\ldots(1)$$

If C m equals the normal pitch, this equation becomes

$$N = 2\pi \times \text{tan. } \phi \times 1 \ldots\ldots\ldots\ldots\ldots\ldots(2)$$

Let $\phi = 75°$, and C m = the normal pitch, then

$$\begin{aligned} N &= 2\pi \times 3\cdot 732 \times 1 \\ &= 23\cdot 45 \text{ nearly} = \text{say } 24, \end{aligned}$$

which is the first whole number above the obtained value; that is, 24 is the smallest number of teeth which can be employed in a wheel which is to have arcs of approaching and receding contact each equal to the pitch.

By increasing the angle of obliquity, and by decreasing the length of path of contact, the number of teeth in the smallest wheel of a set may be considerably decreased.

For example: Let $\phi = 70°$, and the half path of contact C $m = \frac{3}{4}$ the normal pitch, then

$$\begin{aligned} N &= 2\pi \times \text{tan. } \phi \times n \\ &= 2\pi \times 2\cdot 747 \times \cdot 75 \\ &= 12\cdot 94 \text{ nearly} = \text{say } 13. \end{aligned}$$

301. The amount of clearance and back-lash need not be so great as shown in the figure; both of these may be varied by varying the distance between the centres.

For equal arcs of approach and recess the faces of the teeth of the smaller wheel must be longer than those of the larger; therefore, if the lengths of the faces of the teeth are to be equal, a larger wheel may be employed to drive than to follow; this is a result similar to Case II. of the cycloidal form of tooth.

302. In fig. 278 is shown a rack and pinion in gear; the pinion has 13 teeth of 1 inch pitch, and the angle of obliquity is 20°. The acting surfaces of the teeth of the pinion are involutes, and those of the rack are straight lines at right angles to the path of contact F F, and therefore inclined to the pitch line at 70°. A is the centre of the pinion and *p* C its pitch circle; *p l* is the pitch line of the rack, and F C *l* the angle of obliquity. A D is drawn from A at right angles to F F, then A D is the radius of the base circle R D H.

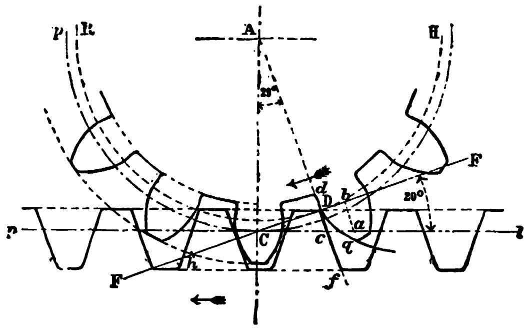


Fig. 278.

The diameter of the pitch circle is to be found by the usual formula—

$$D = \frac{P}{\pi} \times N,$$

which gives in the present case

$$D = 4{\cdot}1366 = 4\tfrac{1}{8} \text{ inches nearly.}$$

C *a* is equal to the circular pitch, and *a b* is drawn at right angles to C *b*, therefore C *b* is the normal pitch. The half-length C D of the path of contact is less than the normal pitch, and therefore, since C *h* = C D, the whole path is

less than twice the normal pitch; therefore the arc of contact will not admit of two pairs of teeth being always in gear.

The length of the path of approach D C for a given wheel, assuming the pinion to be the driver and to rotate in the direction shown by the arrow, may be found as follows:—

We have shown before that C b : C a : : A D : A C. Let θ denote the angle C A D, which is equal to the angle of obliquity, then

$$\text{C D} = \text{A C} \times \text{sine } \theta.$$

In the present case $\theta = 20°$, the pitch C a = 1 inch, and the number of teeth = 13; therefore, substituting the values of θ and A C in this equation, we get

$$\text{C D} = \frac{4 \cdot 1366}{2} \times \cdot 342 = \cdot 7072 \text{ inch nearly.}$$

The normal pitch C b = 1 × cosine θ = ·93969 inch.

$$\frac{\text{C D}}{\text{C } b} = \frac{\cdot 7072}{\cdot 93969} = \cdot 752 \text{ inch.}$$

∴ the arc of contact = circular pitch × 2 × ·752 = 1 × 2 × ·752 = 1·504 inch,

which is a little over $1\frac{1}{2}$ times the circular pitch C a.

The faces and portions of the flanks of the teeth are involutes, as D q, of the base circle R D H; the remaining portions of the flanks are radial lines, as D d.

If the base circle coincides with the pitch circle, the teeth of the rack become straight lines at right angles to the pitch line of the rack; and the path of contact is confined to the pitch line; and therefore the teeth of the pinion touch those of the rack in a point, which point is in the pitch line.

SECTION III.

EXAMPLES OF SPUR AND BEVEL WHEELS—THE ODONTOGRAPH.

303. As examples of the application of the curves described in previous articles, we have selected the following examples which include both Cases I. and II. In the former a rack and pinion are shown, also a pair of bevel wheels in gear; and in the latter a spur wheel and pinion, and a pinion and internal wheel also in gear. In each example the curves

are shown. Approximate method of drawing the curves by means of arcs of circles will be given, the radii of which are obtained from the *odontograph*.

304. Rack and Pinion in Gear.—In figs. 1, 2, and 3, Plate XII., are shown a rack and pinion in gear, the teeth of which are described on the principle stated in Case I. Fig. 2 is a front elevation, fig. 1 a plan, and fig. 3 an end elevation, the top half of which is a cross section, or sectional end elevation, made by the plane S A, fig. 2. The pitch is 1 inch, and the pinion has 16 teeth; therefore D, the diameter of its pitch circle, is, by the equation already given, $= \frac{P}{\pi} \times N = \cdot 3182 \times 16 = 5\cdot 0912 = 5\frac{3}{32}$ inches nearly. The figures are drawn to a scale of $\frac{1}{2}$.

The curves for the tops and bottoms of the pinion's teeth are involutes of the pitch circle, and radial lines respectively of the rack's, cycloids described by a circle A a C, whose diameter is equal to the radius of the pitch circle of the pinion, and lines at right angles to the pitch line $p\,l$ of the rack.

305. Draw the centre lines $a\,x$, $b\,y$ (which contain the projections of the axis of the pinion), and the pitch lines, fig. 2; divide the latter into lengths of 1 inch, and mark off the proportions of the teeth. In the example we have shown a pair of teeth in contact along the line A C; to do this make C m, C n each equal to one-half the thickness W of a tooth, and then from m and n set off the pitch. Having divided the pitch circle and the pitch line for the pitch, set off a tooth for each, as P and R; and draw the curves 0 q, $r\,s$ for the tops of the teeth, by the construction lines already shown. From 0 and r draw the bottoms of the teeth.

We have now to draw a number of curves similar to the portions of 0 q and $r\,s$ which are used for the teeth. For drawing purposes we may substitute for these curves, arcs of circles, whose centres may be found as follows:—The lines 1 I, 2 II, etc., are normals to the curve 0 q at the points I, II, etc., and they are also the *radii of curvature* at I, II, etc., of the curve. Therefore we may substitute for the curve an arc of a circle of a radius t II, which is a mean between the two extreme radii of curvature. In like manner find the

radius of a circle with which to draw the portion of the curve *rs*; the radii of curvature at I, II, etc., are 1—I, 2—II, etc.

If the curves are required for pattern purposes they may be drawn as described in Arts. 314 and 315; or they may be drawn by an actual describing point, according to the definition of the curves given in the preceding articles.

After having obtained the radii of the circles for the tops of the teeth, and having on the pitch lines set off the thickness of the teeth, draw the tops and then the bottoms. The bottoms of the teeth are joined to the cylindrical portion of the wheel by a small curve, the object of which is to strengthen the teeth, as sudden changes of direction, in connected portions of material, tend to weaken the part where the change takes place; and also, in the present case, the *root* of the tooth is increased in size by the curve, and therefore the tooth is stronger than it would otherwise be.

The remaining portions of the figures may be drawn without further instructions, as the constructive lines indicate clearly how the projections are obtained.

On the rack are shown chipping-pieces *c c*. The pinion is connected to its shaft by a key in the usual manner. The shaft is $1\frac{1}{2}$ inch diameter; the dimensions of the key may be taken from Table X., page 165; the width of the teeth is $\frac{1}{2}$ an inch.

306. Spur Wheel and Pinion Gear.—In figs. 1 and 2, Plate XVII., are shown a spur wheel and pinion in gear, the teeth of which are described on the cycloidal principle. Fig. 1 is a front elevation and fig. 2 a plan. The wheel on the axis A is the driver, and that on B the follower; there are 15 teeth in the driver and 38 in the follower; the velocity-ratio is therefore $\frac{38}{15}$. Either A or B may drive, but in the present case we assume A to be the driver. The pitch is $1\frac{1}{2}$ inch, therefore the diameters of the pitch circles are as follows:—

$$\text{For A, } D = \frac{P}{\pi} \times N = \cdot 4774 \times 15 = 7 \cdot 161 = 7\tfrac{5}{32} \text{ inches nearly.}$$

$$\text{For B, } D = \frac{P}{\pi} \times N = \cdot 4774 \times 38 = 18 \cdot 1412 = 18\tfrac{9}{64} \text{ inches nearly.}$$

The diameter of the describing circle is made equal to the diameter of the pitch circle of a pinion of 12 teeth; its dia-

meter is therefore $= \cdot4774 \times 12 = 5\cdot7288 = 5\frac{23}{32}$ inches nearly. The epicycloids for the tops, and the hypocycloids for the bottoms of the teeth, are described by the same rolling circle, which is the constant describing circle for all wheels of the same pitch.

The pinion is solid, with facings on its sides; the wheel has four arms. The wheel and pinion are attached to their respective shafts by means of keys in the usual manner. The teeth of the wheel are connected together by the rim R, which is attached to the boss K by the arms G; upon both sides of each arm are feathers H, which also join the rim and the boss; inside the rim and joining the arms are feathers F; there are also similar feathers F, outside the boss and connecting the arms. The figures are drawn to a scale of $\frac{1}{4}$.

307. Draw the common centre line *c z*, and having calculated the diameters of the pitch circles of the wheels, mark off A *c*, *c* B equal to the radius of the driver and follower respectively, and draw the centre lines *a x*, *b y*, which contain the projections of the axes. From A and B as centres, with A *c*, B *c* as radii, describe the pitch circles; divide these circles into as many parts as there are teeth in the wheels. In the example we have shown a pair of teeth in contact along the line of centres A B. To set out the teeth as shown mark off *c m* on the pitch circle of A, and *c n* on the pitch circle of B, each equal to one-half the thickness of a tooth; then from *m* and *n* commence to divide the pitch circles for the pitch. As the distance taken in the dividers is the cord of the circular pitch, it will of course be a little less than the pitch.

If the number of teeth in the wheel is divisible by 2, divide the pitch circle into halves, and then by trial divide one of these into the required number of parts; if the number of teeth is divisible by a higher number than 2, then divide the pitch circle into such a number of equal parts, and treat one of the parts as in the previous case, and so obtain the division of the part of the pitch circle. Then divide the remaining parts by means of the distance thus obtained, which is, as just stated, less than the circular pitch.

Now draw the circles for the tops and bottoms of the

teeth, and set off a tooth for each wheel, as D and E; draw the curves *o p*, *o q* for tooth of wheel B, and *r s*, *r t* for A. The construction of the epicycloids and hypocycloids is shown in the figure, but for the description we refer the student to the previous sections. For pattern purposes the curves may be drawn as described, or they may be traced out by the actual rolling of the describing circle upon an arc of a circle whose radius is the same as that of the pitch circle; of course, for a pair of wheels four arcs of circles would be required, two for the epicycloids and two for the hypocycloids.

For drawing purposes, arcs of circles may be substituted for the curves, the radii of which may be found in a similar manner to that described for the curves in fig. 1; but we shall give in Arts. 314 and 315 a method of approximating to these curves, which is easier to apply than that just named. The proportions of the several parts are to be taken from the figures, and from the section immediately preceding.

308. Spur, Pinion, and Internal or Annular Wheel in Gear.—In figs. 1 and 2, Plate XIII., are shown an internal or annular spur wheel and a pinion in gear, whose teeth are described on the cycloidal principle. Fig. 1 is a front elevation, and fig. 2 an end elevation or plan, the lower part of which is in section.

The pinion on the axis A is the driver, but, as before stated, either pinion or wheel may drive according to circumstances; there are 12 teeth in the pinion and 38 in the wheel, the velocity-ratio is therefore $\frac{38}{12} = \frac{19}{6}$. The pitch is $1\frac{1}{2}$ inch, therefore the diameters of the pitch circles are $5\frac{23}{32}$ inches, and $18\frac{9}{64}$ inches. The diameter of the describing circle is equal to the radius of the pitch circle of the pinion, as there are 12 teeth in the pinion; and therefore the flanks of the pinion are radial lines. The epicycloids, for the pinion and wheel, and the hypocycloid for the wheel, are drawn as shown by the construction lines; for further description, see Plates III., XI., and XIV., and the text thereon.

The pinion is solid, the wheel consists of a rim R, inside of which are the teeth, and a plate P attached to the rim, K is the boss for the shaft; the rim, plate, and boss are all cast together. Arms are sometimes employed to connect the rim to the boss, instead of a plate, as shown in the example.

309. Draw the common centre line $c\,z$, and having calculated the diameters of the pitch circles, mark off B C equal to the radius of the wheel; and from C, towards B, mark off C A equal to the radius of the pinion. Through A draw the centre line $a\,x$, and draw $b\,y$ parallel to $a\,x$; these lines contain the projections of the axes. The construction lines, with the explanations already given, show how to complete the figures. It must be observed that the tops of the teeth of the wheel are nearer the centre than the bottoms, which is the reverse of the previous example.

310. Pair of Bevel Wheels in Gear.—Figs. 1 and 3, Plate XV., represent the pair of bevel wheels shown in outline in Plate X. A has 24 teeth, and B 20, $1\frac{1}{2}$ inch pitch. The teeth are of the form described as having radial flanks. The projections of the axes are marked $a\,a$, a', and $b\,b$, $b'b'$. Draw the centre lines $a\,x$, $b\,b$, and $b'b'$, and the projections of the pitch circles as shown. Determine the points e, l, n, and o, and draw lines for the tops and bottoms of the teeth (see text for Plate X.). Draw E L, fig. 2, parallel to and at a convenient distance from $e\,l$; produce C f to meet E L in F, and draw e E, l L parallel to it. From E and L as centres, with radius E F, L F respectively, describe arcs of circles G F Q, K F R; these arcs of circles are the boundaries of the portions of the developments of the cones $e\,f\,g$ and $l\,f\,k$, upon the surfaces of which the teeth are set out.

Draw the teeth, fig. 2, as if G F Q, K F R were the pitch circles of a pair of spur wheels of $1\frac{1}{2}$ inch pitch, E and L being their centres; the describing circles being equal to the radii of these pitch circles. The curves c F, d F are shown more fully in Plate XIV. (Plates XIV. and XV. must be studied together, or the student will not understand: the figures on these two Plates are numbered from 1 to 10).

Fig. 5, Plate XIV., shows the frusta of the cones C$f\,g$, C$f\,k$, which are the pitch surfaces of the wheels. Fig. 7 is a development of the cone C$f\,g$; the epicycloid 0...d, which forms the tops of the teeth of the wheel A, is described by the circle F s L (figs. 2 and 6).

Fig. 6 is a development of the cone C$f\,k$; the epicycloid o...c, which forms the tops of the teeth of the wheel B, is described by the circle FrE (figs. 2 and 7). The bottoms of

the teeth are radial lines. Having drawn the teeth, as shown in fig. 2, commence and draw their projections in figs 1 and 3; see Plate XV.

Figs. 8, 9, and 10, Plate XIV., show portions of the wheels, drawn to a scale of $\frac{1}{2}$, with the construction lines required to project the teeth of the smaller wheel. Draw the centre and the pitch lines of fig. 8, corresponding to fig. 1; and draw the teeth, fig. 10, in their true form, as shown in fig. 2. Divide the pitch circle $f r'$, fig. 8, into 20 equal parts, as we are taking the smaller wheel for the example, and set off on the pitch circle the thickness of the teeth, equal to the thickness of the teeth on the pitch circle in fig. 10. Now project lines from t and h, fig. 8, and describe circles for the tops and bottoms of the teeth; make the thickness of the tops and bottoms, measured upon these circles, of the same size as those of the corresponding wheel in fig. 10. That is to say, $u\ v$, $w\ x$, $y\ z$, fig. 8, equal respectively $u\ v$, $w\ x$, $y\ z$, fig. 10; but the projected length $w\ y$, fig. 8, is less than the true length $w\ y$, fig. 10.

From $w\ x$ and $u\ v$ (y and z being in the latter lines), draw lines to o, the centre of the wheel, meeting the circles which form the teeth of the inside of the wheel, and so obtain the projection of those teeth as shown. The curved lines for the tops of the teeth are projections of those in fig. 10, and if the latter are arcs of a circle, then those in fig. 8 may be assumed to be arcs of a circle. The approximate method of describing the teeth of spur wheels, previously given, applies also to the teeth of bevel wheels, as set out in figs. 2 and 10.

The teeth of the wheel B, fig. 9, are to be projected from those of fig 8, as shown by the construction lines drawn from the tooth P. From $w\ x$ draw lines meeting $t\ t$ in $w'x'$; and from $u\ v$ draw lines meeting $f\ r$ in $u'v'$; join $u'v'$ to l, meeting $h\ h$ in $y'z'$. Then $y'u'w'x'v'z'$ is the required projection of the outside of a tooth; from each of these points, or from as many of them as are required, draw lines to C, meeting the lines $h'h'$, $m\ m$, $t't'$, and so obtain the projection of the whole tooth. The remaining teeth are to be drawn in a similar manner, so also are the teeth on the wheel A.

The teeth of the wheel A, fig. 9, are projected from fig. 8,

and those of B are projected from figs. 8 and 10; the projections of B in figs. 1 and 3 being similar.

To facilitate the drawing of such projections as figs. 1, 3, 8, and 9, a fine needle may be fixed at C a', α, and e for the set-square to bear against.

The teeth of all bevel wheels, whose axes are in the same plane, are projected in the manner just described; but, of course, if the flanks are not radial, then their true form must be drawn in fig. 10, and then projected in a similar manner to the tops of the teeth.

311. Willis's Odontograph. — Professor Willis invented in 1838 an instrument to facilitate the drawing of approximate cycloidal teeth. The instrument is called by its inventor the *Odontograph;* fig. 279 is a drawing of it to a scale of $\frac{1}{2}$, and in the following article a description is given, illustrated by examples. It is usual in practice, as for example in setting out the teeth of a *pattern* or *model*, to substitute arcs of circles for the portions of the cycloidal curves required; and, providing the proper radii are employed, the errors introduced by the substitution may, for practical purposes, be overlooked as inappreciable. The radii to be employed are the *mean radii of curvature* of the cycloidal arcs; these can be found by calculation or by construction, but as the operation would be required for every wheel, it is more convenient to calculate them beforehand, and arrange them in a table. This has been done by Professor Willis, and such a table is given on the drawing of his odontograph; the table is calculated by assuming the smallest wheel in a set to have 12 teeth.

The angles which these mean radii of curvature make with the pitch line have been fixed by the inventor of the odontograph at 15°, as that angle gives the best form to the teeth. And the point at which the approximate curves, forming the top of the tooth of one wheel and the bottom of the tooth of the other, are exactly at the distance of half the pitch from the point where the line of centres cuts the pitch lines.

312. In fig. 280, p C E is the pitch circle of a wheel, the arcs C D, C E are each equal half the pitch, and D A, E A are radii of the pitch circle. B h, P k are drawn through D and E respectively, making the angles B D A, k E A each

THE ODONTOGRAPH

TABLES SHOWING THE PLACE OF THE CENTRES UPON THE SCALES.

CENTRES FOR THE FLANKS OF TEETH

Number of Teeth.	Pitch in Inches: 1	1¼	1½	1¾	2	2¼	2½	3
13	129	160	193	225	257	289	321	386
14	69	87	204	221	139	156	173	208
15	49	62	74	86	99	111	123	148
16	40	50	59	69	79	89	99	191
17	34	42	50	59	67	75	84	101
18	30	37	45	52	59	67	74	89
20	25	31	37	43	49	56	62	74
22	22	27	33	39	43	49	54	65
24	20	25	30	35	40	45	49	59
26	18	23	27	32	37	41	46	56
30	17	21	25	29	33	37	41	49
40	15	18	21	25	28	32	35	42
60	13	15	19	22	25	28	31	37
80	12	...	17	20	23	26	29	35
100	11	14	...	...	22	25	28	34
150	...	13	16	19	21	24	27	32
Rack	10	12	15	17	20	22	25	30

CENTRES FOR THE FACES OF TEETH.

Number of Teeth.	1	1¼	1½	1¾	2	2¼	2½	3
12	5	6	7	9	10	11	12	15
15	...	7	8	10	11	12	14	17
20	6	8	9	11	12	14	15	18
30	7	9	10	12	14	16	18	21
40	8	...	11	13	15	17	19	23
60	...	10	12	14	16	18	20	25
80	9	11	13	15	17	19	21	26
100	...	...	...	...	18	20	22	...
150	...	...	14	16	19	21	23	27
Rack	10	12	15	17	20	22	25	30

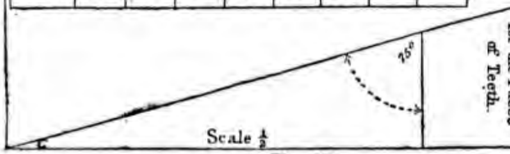


Fig. 279.

equal to 75° (90° − 15°); these lines are normals to the cycloidal curves, and the mean radii of curvature lie in them.

The distances D *h* and E *k* are obtained from the tables on the odontograph, as will be explained presently, then *k* C and *h* C are the respective radii with which to describe the faces and flanks of the teeth. *k* P is the radius of curvature at the point P of the *epicycloidal* arc C P *m*, and *h* B is the radius of curvature at B of the *hypocycloidal* arc C B *n*. If these arcs are small, then the error introduced by substituting arcs of circles for them will also be small.

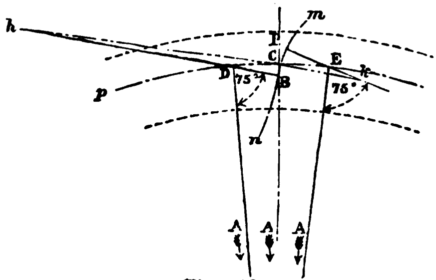


Fig. 280.

313. For practical use the odontograph may be made of cardboard or sheet metal; the following description, and fig. 279, will enable the student to make one for his own use:—Having provided a rectangular piece of cardboard, 13 inches by 7½ inches, upon one of its long edges, as A B, take a point T, about 2½ inches from B; on each side of T set off distances of ½ an inch, divide each of these into ten equal parts, and number them as shown; from T draw T C, so that the angle B T C contains 75°. Fig. 279 is drawn to a scale of ½, and therefore the distance contained by each of the ten divisions is only ¼ an inch; also the upper scale only extends to 165 divisions, whereas in the original there are 200.

The tables are to be copied from the figure; the top one contains the numbers for the centres of the flanks, and the bottom one those for the faces of the teeth. The first column in each table contains a list of certain wheels of from 13 to 150 teeth; the whole of the wheels are not given, because the error in taking from the column the numbers for the

wheel nearest to the one required is very small and practically inappreciable. The remaining columns contain respectively the numbers for the centres of the faces and flanks of the teeth of the wheels given in the first column, for pitches advancing by ¼ inches from 1 inch to 2½ inches, and for a pitch of 3 inches. The numbers for intermediate pitches may be found by direct proportion from those given; thus, for ½ inch pitch, by halving those of 1 inch pitch; for 3½ inches pitch, by doubling those of 1¾ inch pitch; and so on for other pitches.

314. The following examples will explain how to use the odontograph :—

I. *To draw the Teeth of Spur Wheels.*—Figs. 1 and 2, Plate XVI., show a spur wheel of 29 teeth, 1¼ inch pitch; A is its centre, and P C its pitch circle. Fig. 1 is drawn to a scale of ¼, and fig. 2 shows a portion of the former figure drawn full size.

From A draw any radial line A B, fig. 2, meeting the pitch circle in *m*; from *m* set off along the pitch circle *m* D, *m* E, each equal to one-half of the pitch; from D, E, draw radial lines D A, E A.

For the flank of the tooth place the line C T of the instrument upon the line A D, so that T coincides with D; now look to the tables of *centres for the flanks of teeth*, and in the column of 1¼ inch pitch, opposite 30 teeth (nearest to 29) is the number 21. The point numbered 21, counting from T to A, on the *scale of centres for the flanks of teeth*, is the centre required; we will call the point *h*; and from *h* as a centre with a radius *h m* describe the arc *m p*, which is the required flank.

To describe the face, place the line C T upon A E; in the table of *centres for the faces of teeth*, opposite 30, will be found the number 9, counting from T to B; mark off this number from the *scale of centres for the faces of teeth;* we will call the point *k*; from this point as a centre with a radius *k m* describe the arc *m n*, which is the required face. By repeating these curves the other side of the tooth can be drawn, and also the remaining teeth. From A as a centre with radii A *h*, A *k*, describe circles *h q*, *r k*. These circles contain the centres (*h*, *k*) of all the teeth, which are to be described with the radii *h m*, *k m*.

II. *To draw the Teeth of Racks.*—Fig. 281 shows a portion of a rack of 1 inch pitch, drawn full size. The centres for the faces and flanks of the teeth are obtained in a similar manner to those in the previous example; *p l* is the pitch line; *m* is any point in the pitch line; and *m* D, *m* E, are each equal one-half the pitch, measured along the pitch line. D A and E A are drawn at right angles to the pitch line, because *p l* is a straight line, which may be considered as a circle of infinite radius, and D A, E radii of that circle. To draw the flank *m p*, place the line T C of the instrument so that T coincides with D, and T C with D A; and mark off the point 10. Let *h* denote this point, then *h m* is the radius required. The point *k* is found in a similar manner, always noticing that *h* and *k* are on opposite sides of *m*. In the case of racks *h m* and *k m* are equal, because the curves we are approximating are equal and similar cycloids; and therefore the numbers in the bottom line of each table in fig. 279 are equal for the same pitch.

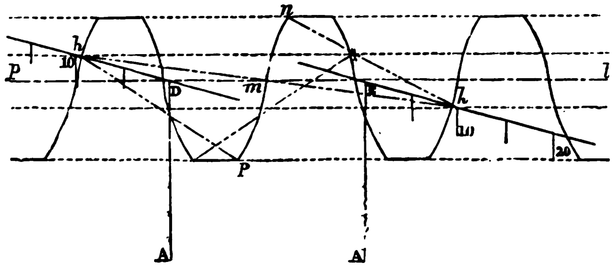


Fig. 281.

315. In figs. 1 and 3, Plate XVI., are shown a spur wheel and pinion in gear, whose teeth are described by means of the odontograph. Fig. 1 is a front elevation, and fig. 3 an end elevation, a portion of which is in section, as made by a plane passing through *b* B, fig. 1. The pinion A is the driver, and the wheel B the follower; the former turns in the direction indicated by the arrow. A has 19 teeth, and B has 50, $1\frac{1}{4}$ inch pitch. The diameters of the pitch circles are by the equation (3), for A, $7{\cdot}558 = 7\frac{9}{16}$ inches nearly, and for B $19{\cdot}890 = 19\frac{7}{8}$ inches nearly.

Draw the centre line $a\,y$, and fix upon a point for the centre of one of the wheels, say A, for the pinion; from A mark off along $y\,a$, $A\,b = \frac{1}{2}$ of $7\frac{9}{16}$ inches, and from b mark off $b\,B = \frac{1}{2}$ of $19\frac{7}{8}$ inches; through A and B draw the centre lines $d\,x$, $e\,z$, which contain the projections of the axes. From A and B as centres, with radius $A\,b$ and $B\,b$ respectively, describe the pitch circles P C, P C, and circles for the tops and bottoms of the teeth, taking the proportions for these and also those for W and R from Art. 244, p. 198. The teeth are to be described as previously shown; the centres h and k are taken from the tables for $1\frac{1}{4}$ inch pitch, fig. 267. The numbers for the faces are for A, 8, and for B, 10; which are the numbers for wheels whose numbers of teeth are nearest to the required ones. Those for the flanks are taken between 37 and 31 for A, and between 18 and 15 for B; which are the numbers for wheels of 18 and 20, and 40 and 60 teeth respectively.

The pinion is solid, with facings on each side; $\frac{3}{16}$ inch thick, with 4 inches radius, and $3\frac{1}{2}$ inches through.

The wheel has six arms of a cross section shown at X, fig. 3, as made by the plane S_1P_1; they are connected to the rim R and boss K by feathers F, F; on both sides of each arm are feathers H, which also join the rim and the boss, the whole being cast together. The boss is 5 inches diameter, and 4 inches through; the whole is 20 inches diameter. For the remaining dimensions, see the figures and tables.

TABLE XVII.

Proportion of Spur and Bevel Wheels.

The pitch	(=1·25″)	$=p$.
Top of the tooth, T		$=p \times$ ·33.
Bottom of the tooth, B		$=p \times$ ·42.
Total depth of tooth, T+B		$=p \times$ ·75.
Thickness of tooth on pitch line, W		$=p \times$ ·45.
Space between teeth on pitch line, S		$=p \times$ ·55.
Thickness of rim, R		$=p \times$ ·45.
Thickness of arms, G		$=p \times$ ·45.
Width of feathers, F, F		$=p \times$ ·45.
Thickness of feathers, H, F		$=p \times$ ·45.
Thickness of boss round shaft, K		$=p \times 1$.
Usual width of teeth, L		$=p \times 2·5$.

316. The inappreciable error which we referred to in a previous article, as arising from the use of numbers in the tables, fig. 279, corresponding to a wheel of a different number of teeth to the required one, is noticed by the inventor of the instrument in the following manner:*—

"It is unnecessary to have numbers corresponding to every wheel, for the error produced by taking those which belong to the nearest as directed, is so small as to be inappreciable in practice. I have calculated the amount and nature of these errors by way of obtaining a principle for the number and arrangement of the wheels selected. It is unnecessary to go at length into these calculations, which result from very simple considerations, but I will briefly state the results.

"The difference of form between the tooth of one wheel and that of another is due to two causes: (1) the difference of curvature, which is provided for in the odontograph by placing the compasses at the different points of the scale of equal parts; (2) the variation of the angle D A E, fig. 4, Plate XVI., which is met by placing the instrument upon two radii in succession.

"The first cause is the only one with which these calculations are concerned. Now in a 3 inch pitch the greatest difference of form, produced by mere curvature in the portion of tooth which lies beyond the pitch circle, is only ·04 inch between the extreme cases of a pinion of twelve and a rack, and in the acting part of the arc within the pitch circle is ·1 inch; so that, as all the other forms lie between these, it is clear that if we select only four or five examples for the outer side of the tooth, and ten or twelve for the inner side, that we can never incur an error of more than the $\frac{1}{200}$th of an inch in 3 inch pitch, by always taking the nearest number in the manner directed; and a proportionably smaller error in smaller pitches. But to ensure this, the selected numbers should be so taken, that their respective forms shall lie between the extremes at equal distances. Now it appears that the variation of form is much greater among the teeth of small numbers than among the larger ones, and that in fact the numbers in the two following series are so arranged, that

* Willis's *Principles of Mechanism*, page 139.

the curves corresponding to them possess this required property:—

For the outer side of the tooth—12, 14, 17, 21, 26, 34, 47, 73, 148, Rack.

For the inner side of the tooth—12, 13, 14, 15, 16, 17, 19, 22, 26, 33, 46, 87, Rack.

"Now these numbers, although strictly correct, would be very inconvenient and uncouth in practice if employed for a table like that in question, when convenience manifestly requires that the numbers, if not consecutive, should always proceed either by twos or fives, or by whole tens, and so on. They are only given as guides in the selection, and by comparing them with the actual table, their use in the formation of the first column will be evident."

SECTION IV.

STRENGTH OF TEETH.

In designing the teeth of wheels we have to consider, in addition to the points already mentioned, *their strength*, that is, the power they are capable of transmitting under given conditions. The two extreme cases are—(*a*) a wheel revolving at a high rate of speed; (*b*) and a wheel at rest. In the first case we have to take into consideration the velocity of rotation; in the second, simply to determine the absolute or ultimate strength of the teeth. The latter being the simpler case we consider it first, merely observing that the substance of which a wheel is composed materially affects its strength. Cast-iron is chiefly employed for machine and mill gearing; hence our calculations will be confined to that metal. Steel, brass, and malleable cast-iron are employed in special cases.

317. Strength of Teeth when at Rest.—The most common case we can mention of a pair or of a train of wheels at rest under the action of forces, is that of a loaded crane. When the raising of the ultimate weight is effected, and the rotation of the wheel ceases, the strain upon the teeth is a *dead* strain or load, and it is the greatest load which the teeth should be calculated to carry.

Let fig. 282 represent a pair of wheels in gear, with centres A and B, and let C and D be a pair of teeth in contact, the point of contact being at e, the extremity of the tooth D; and, further, let the wheel A be the driver; the strain upon the tooth D is similar to the strain upon a beam fixed at one end and loaded at the other; hence the length $e\,f$ of the tooth represents the projecting beam or cantilever, and the forces transmitted by C represent the load. We have already seen how the strength of a beam under the foregoing conditions varies; therefore treating a tooth in a similar manner, we can calculate its strength; the strength of the individual tooth will be its ultimate resistance to fracture.

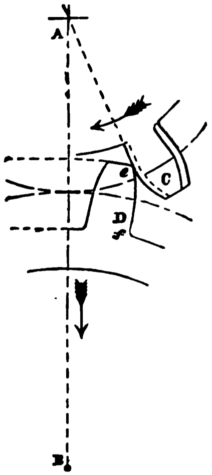


Fig. 282.

Let fig. 283 represent a tooth of $2\frac{1}{2}$ inch pitch fixed at b, and loaded at a. We shall assume the thickness at the neck b to be equal to that in the pitch; then from the proportions formerly given the length is $1\frac{7}{8} = 1{\cdot}875$ inches three quarters of the thickness; d is $1\frac{1}{8} = 1{\cdot}125$ inches; and the width at b is $2\frac{1}{2}$ times the pitch,

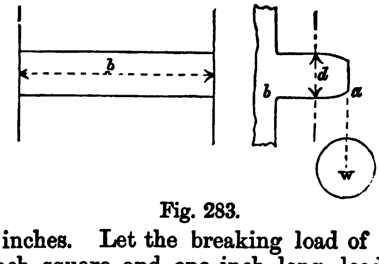


Fig. 283.

that is, $6\frac{1}{4}$ inches. Let the breaking load of a bar of cast-iron, one inch square and one inch long, loaded as in the figure, be 6000 lbs., then we have generally, where W is the breaking load,

$$W = \frac{d^2 \times b \times 6000}{l} \text{ ....(1.)}$$

Substituting the given values of a $2\frac{1}{2}$ inch pitch tooth, we get

$$W = \frac{1{\cdot}125^2 \times 6{\cdot}25 \times 6000}{1{\cdot}875} = 25312{\cdot}5.$$

The value of W thus obtained must be multiplied by the factor of safety to obtain the safe working load. If we assume the factor of safety to be $\frac{1}{10}$th, then

$$\text{the safe working load} = \frac{W}{10} = \frac{25312}{10} = 2531 \text{ lbs.}$$

In the example just given, we have assumed that only one pair of teeth are in gear, but if there are two pairs in gear at the same instant, then we must multiply the safe working load, as found from the above formula, by 2, when it becomes $2531 \times 2 = 5062$ lbs. A small value for the depth d has been taken, but of course this only affects the actual example taken, as in every case the actual values must be substituted in the equation.

We have assumed in this last case that the load is distributed equally across the whole width of the tooth, but there are cases where, through wear and other causes, the teeth are subjected to severer strains, through the stress coming upon part of the tooth only, or even upon the corners. In fig. 284, if we consider $a\,b = a\,c$, then the greatest stress will be near the line $c\,b$. Assuming the pressure to act at a, the following formula expresses the strength of the tooth; it was originally given by Tredgold:—

$$W = \frac{f d^2}{5} \;.........(1)$$

where, as before, d is the thickness of the tooth, and f depends upon the material. Making an allowance by subtracting $\frac{1}{3}$ for wear, the equation becomes

$$W = \frac{f d^2 \, (1 - \frac{1}{3})^2}{5} = \frac{f d^2}{11{\cdot}25}.$$

For cast-iron f is taken as 15300; if this be substituted in the above equation,

$$d = \sqrt{\frac{W}{1360}} \text{ (nearly)};$$

or in words, the thickness of a tooth in inches should be equal to the square root of the stress on the tooth in pounds, divided by 1400.

By an extraordinary error in writing $\frac{fd^2}{10 \cdot 25}$ for $\frac{fd^2}{11 \cdot 25}$, our best text-books give 1500 instead of 1400.

We have here shown what the bending moment is, arising from the total force transmitted by a pair of wheels, considering that this force acts on one pair only.

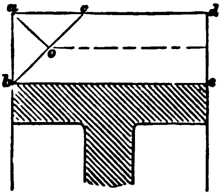


Fig. 284.

In fig. 284, the cross section of the rim of the wheel is seen, while $a\,d\,e\,b$ is the face of the tooth upon which the stress should come evenly; but if it acts on one corner a, we must consider $c\,b$ as a plane of section, and $a\,o$ as a perpendicular from a. Let d be the thickness of the tooth, and $c\,b = b$, and the perpendicular $a\,o = l$, then according to Tredgold, Rankine, etc., the greatest stress upon the section, where P is the force, is

$$p = \frac{6 P l}{b d^2}$$

which attains its greatest value when the angle $a\,c\,b$ is 45°, and $b = 2\,l$, under which conditions we obtain

$$p = \frac{3 P}{d^2} = f;$$

$$\therefore \text{ thickness for tooth} = d = \sqrt{\frac{3 P}{f}},$$

for cast-iron f will be 4500 lbs. on the square inch.

318. In calculating the strength of the teeth of bevel wheels, the mean size of the teeth must be taken.

The strength of teeth must be considered in another aspect, namely, the velocity at which the wheels are moving. While the pressure on a tooth varies directly as the horse-power transmitted, it varies inversely as the velocity of revolution. If one wheel transmits 10 horse-power, and a second 20 horse-power, both moving at the same velocity, the strain on that carrying 20 horse-power will be double the other. Again, if one wheel moves with a velocity of 50 feet per minute, and the second at 200, both transmitting the same horse-power,

the strain on the latter will be only one quarter that on the first.

If H is the horse-power transmitted, v the velocity in feet per second, then the pressure in pounds on the wheels is given by the formula—

$$P = \frac{550 \times H}{v}.$$

This may be illustrated by the following question:—Suppose a fly wheel 35 feet in diameter makes 30 revolutions per minute, works a pinion 6 feet in diameter, and transmits 250 horse-power; find the statical pressure on the teeth.

$$\text{Velocity per second} = \frac{35 \times 3{\cdot}1416 \times 30}{60} = 55 \text{ feet (nearly).}$$

$$P \text{ (pressure)} = \frac{550 \times 250}{55} = 2500 \text{ lbs.};$$

that is, the pressure on the teeth would be 2500, when 250 horse-power are transmitted at the above velocity.

SECTION V.

MORTICE WHEELS—FLANGED WHEELS—HOOKE'S GEARING—ROBERTSON'S FRICTIONAL GEARING.

319. Mortice Wheels.—It is usual in mill gearing to employ wooden teeth or *cogs* for one of a pair of wheels; the wheel which has these wooden teeth is called a *mortice wheel*, as the teeth are inserted in mortices in the rim of the wheel.

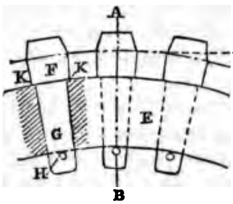


Fig. 285.

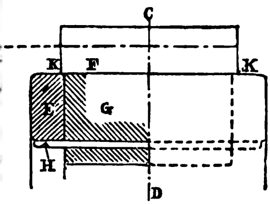


Fig. 286.

Figs. 285 and 286 represent a portion of a mortice wheel; E is the rim, F G the tooth or cog, the *shank* is G, which

fits the mortice in the rim; H is an iron pin, usually a piece of iron wire, which passes through the shank just inside the rim, and prevents the cog from working loose and leaving the mortice. The letters refer to both figures.

On the left-hand side of fig. 285 the rim is in section, showing the shank and the mortice in the rim. On the left-hand side of fig. 286 the rim and a portion of the shank are in section, showing the mortice and the connecting pin H. The shank is made smaller than the portion outside the mortice, so that there is a shoulder K K on each side of the cog which bears upon the rim.

320. Wooden teeth moving at slow velocities, or at rest, are not so strong as iron ones; it is therefore necessary to make the teeth of mortice wheels thicker than those of the iron with which they work. A common proportion is ·6 of the pitch for the wooden cogs, and ·4 of the pitch for the iron teeth, little or no play being allowed, as the teeth of such wheels are always carefully trimmed before they are used.

For a given pitch, the teeth of a mortice wheel and those of the iron one in gear with it are shorter than those of ordinary iron wheels, because the proportions are those due to a reduced thickness of tooth; thus the thickness, measured on the pitch line, of the iron tooth is only ·4 of the pitch, whereas in ordinary gearing it is ·45 of the pitch. Therefore the length of the tooth should be reduced in the proportion of $\frac{\cdot 4}{\cdot 45} = \cdot 88$ of the usual length.

The mortice wheel is usually the follower, but not always; the practice is to make the smaller wheel with iron teeth. The woods employed for cogs are the following:—Hornbeam, crab-tree, locust, apple-wood, holly, and beech; the first is the one most commonly used.

321. Flanged Wheels.—The strength of the teeth of wheels may be considerably increased by *flanging* or *shrouding* them. In figs. 287 and 288 is shown a portion of a flanged wheel; F is the rim, and E, E the flanges. The flange consists of a ring on the face of the wheel passing from the inner edge of the rim to within a short distance of the pitch line, the whole being cast together. The result is that the teeth have their strength increased by having their ends supported by the

flange, and thus in a sense having their length reduced by adding an additional amount of metal.

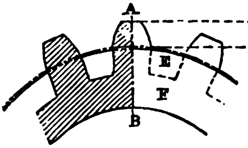


Fig. 287.

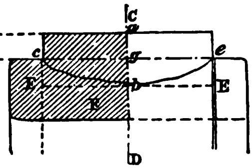


Fig. 288.

If we suppose a tooth of a flanged wheel to be broken, the line of fracture would be somewhat similar to the irregular line $c\,b\,e$. Let $a\,b =$ the length of a tooth, $g\,b$ the flank, and let the flanges come up to the pitch line, then we may take the mean length of the flank, plus the length of the face, for the length of the tooth in our calculations.

322. As an example, let the pitch be $2\frac{1}{2}$ inches, then we have

$$l = a\,g + \frac{g\,b}{2} = \cdot 35\,p + \frac{\cdot 40\,p}{2} = \cdot 55\,p = \cdot 55 \times 2\cdot 5 = 1\cdot 675 \text{ inch;}$$

and by equation (1),

$$\mathrm{W} = \frac{d^2 \times b \times 6000}{l}$$

$$= \frac{1\cdot 125^2 \times 6\cdot 25 \times 6000}{1\cdot 675} = 28335 \text{ lbs. (nearly).}$$

In the example given, where the teeth are not flanged, we have W = 25312 lbs., so that the ratio of the strength of flanged and ordinary teeth is as, say, 283 : 253; or the flanged tooth is about 12 per cent. stronger than the other.

323. Hooke's Gearing.—We have already studied the various points respecting the action of the teeth of a pair of wheels in gear, as to form and length of path of contact, and the strength of the teeth. Therefore we can now consider how these quantities vary under given conditions. If we increase the number of teeth in a pair of wheels we get a smoother action, but we reduce the strength of the teeth, and consequently reduce the power they are capable of transmitting; again, if we increase the length of the path of contact, we also reduce the strength of the teeth, as that augmentation involves

an addition to the length of the teeth, which corresponds to an increase of leverage or breaking strain.

The arrangement invented by Dr. Hooke combines the advantages of smoothness of action afforded by a reduced pitch, and of strength due to the original pitch.

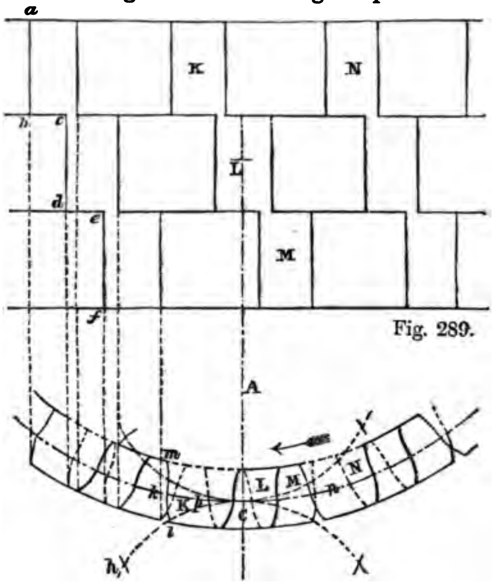


Fig. 289.

Fig. 290.

In figs. 289 and 290 is shown a portion of a wheel with Hooke's *stepped* teeth; fig. 289 is a plan and 290 an elevation; in this figure we have shown the projections of the fronts of the teeth only. The width of the wheel is divided into a number of plates, as K, L, M; and instead of the teeth in these plates forming a straight line parallel to the axis, as in an ordinary spur wheel, they form a series of steps *a b*, *c d*, *e f*. We have now to consider what is gained by such an arrangement.

324. The length of the path of contact depends upon the length of the faces of the teeth. In the present case we

shall refer to receding contact only, and shall assume the teeth to be cycloidal.

Let the pitch P be $1\frac{1}{2}$ inch, and let h C be the describing circle. Then, if the arc of receding action is to be equal to the pitch, and if the arc C h equals the pitch, the circle containing the extremities of the faces of the teeth of the wheel A must pass through h; but if the path of action is only to equal one-half the pitch, then that circle must pass through l, C l being one-half C h.

Let $k\,n$ equal the pitch, and let the wheel rotate in the direction shown by the arrow. The tooth K, whose front is marked $l\,k\,m$, is just ending contact, having passed through an arc C k, equal one-half the pitch; but before contact has ended the tooth L comes into contact, and before L has ended contact the tooth M comes into contact; contact finally ending for the last tooth M at l.

In the figures the front of M is a little on the right of the centre line, so that by the time M ends contact the arc of receding contact will be a little more than twice C k; that is, a little greater than the pitch. This increase in the length of the arc of action has been produced by the stepping of the teeth.

The thickness of the teeth is that due to the pitch $k\,n$; by this arrangement we therefore get a short strong tooth. The chief objection to the arrangement is the increased difficulty in obtaining correct castings, so that each tooth in each plate shall successively come into contact at the proper time; but this difficulty, if it really exists, can be overcome by good workmanship.

Messrs. W. Collier & Co., Salford, have for many years employed stepped teeth for the rack and pinion motion for driving the tables of their planing machines, and with complete success.

325. The number of plates and the amount of step can be varied at pleasure; we will see how these vary with the other conditions, as length of face of tooth and length of path of contact. Let P denote the pitch proper; that is, the distance, measured on the pitch line, from the outside of one tooth to the outside of the next, as $k\,n$ of the two teeth K N; let p denote the reduced pitch; that is, the distance

from the outside of a tooth in one plate to the outside of the adjoining tooth in the next plate, as $k\,p$ of the two teeth K, L; and let n denote the number of plates. Let the fronts of the two teeth N and M, in the outside plates, be in the same line; then the pitch $p=\frac{P}{n-1}$. Let $n=4$, and $P=1\frac{1}{2}$ inch. Then

$$p=\frac{P}{n-1}=\frac{1\cdot 5}{4-1}=\frac{1\cdot 5}{3}=\cdot 5 \text{ inch.}$$

The length of the path of contact, and the length of the faces of the teeth, are those due to this reduced pitch. In practice it is usual to allow the teeth in each plate to overlap those in the adjoining ones, so as to get a stronger arrangement; if, therefore, there are only four plates, and they are arranged as just stated in the example, the teeth would not overlap sufficiently.

A better arrangement would be got by making $p=\frac{P}{n}$, so that if

$$n=4, \text{ and } P=1\cdot 5 \text{ inch, } p=\frac{1\cdot 5}{4}=\cdot 375 \text{ inch;}$$

and the tooth M in the last plate will be ·375 inch in advance of the tooth M in the first plate. The other variations, resulting from a fixed length of arc of contact, of a given length of face of tooth, etc., may easily be worked out by the student.

We have spoken of plates in describing the foregoing arrangement; in practice these plates are all cast together, and it is the correct arranging of the plates of each wheel of a pair that causes the difficulty in construction to which we previously alluded.

326. Robertson's Frictional Gearing.—The use of friction wheels has become more common since the introduction of Mr. Robertson's system. In figs. 291 and 292 is shown a portion of a pair of spur wheels having angular grooves on the outer edges of their rims, as in the system named. Fig. 291 is a front elevation showing the pitch circles p C, p C; and fig. 292 is an end elevation, the left-hand half of which is in section.

The angle of the grooves is 40°, and the lengths of the tops of the projecting ridges of each wheel are equal on each side of the pitch line p' C′. It is stated by the inventor of such gearing that, after being wrought for some time together and thoroughly cleaned and fitted, the bite or hold of the surface is fully $1\frac{1}{2}$ times the force exerted in holding them in contact; and in practice, where the wheels are not less than 2 feet diameter, a pressure equivalent to the power required to be transmitted is all that is wanted to retain the wheels in contact, and to secure the necessary adhesion without slipping. Also, that a pressure of $1\frac{1}{2}$ ton is sufficient to transmit one hundred indicated horse-power, with wheels working at a speed of one thousand circumferential feet per minute.

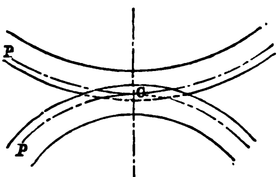


Fig. 291.

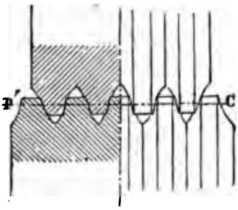


Fig. 292.

This system is applicable to bevel wheels as well as to spur wheels, both external and internal.

SECTION VI.

EXAMPLES: SCREW WHEELS—WORM AND WORM WHEELS.

In the following articles we shall work out examples of the screw wheels, previously described. For the first example we shall take the simple case, that of a pair of equal and similar wheels called spur mitre wheels; and for the second example, the case of a pair of unequal wheels. After these we give examples of worms and worm wheels.

327. Pair of Equal Screw Wheels in Gear.—In figs. 293 and 294 are shown a pair of equal screw wheels in gear, and in figs. 1-6, Plate XXI., one of these wheels is shown in

detail; to these figures we now refer. These wheels have 12 right-handed teeth or threads, and as they are equal and similar, and their axes are at right angles, the angles of inclination of the threads are equal in each wheel, so that the angle of inclination is 45°. The circular and the axial pitches are therefore equal; that is, the circumference of the pitch circle and the pitch of the helix are equal; in the example each is 12 inches.

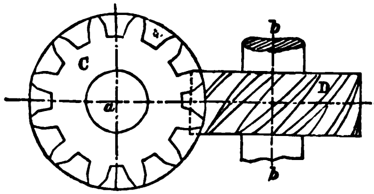


Fig. 293.

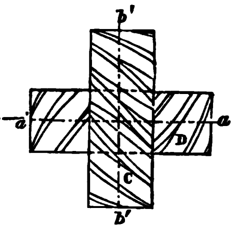


Fig. 294.

The divided axial pitch $= \frac{\text{axial pitch}}{12}$; and the divided circular pitch $= \frac{\text{circular pitch}}{12}$; therefore each of the pitches $= \frac{12}{12} = 1$ inch.

Fig. 1 is an elevation projected upon a plane at right angles to the axis, the projections of which are marked *a*, *a′a′*; fig. 2 is an elevation on a plane at right angles with that on which fig. 1 is projected; that is, on a plane parallel with the axis. On the left of fig. 2 is shown a portion of the cylinder out of which the wheel has been made, and the whole length B VI is 6 inches long, just half the pitch.

The cylinder 0 6 VI D, shown in dotted lines, is the pitch cylinder, and the curved line 0...VI is one-half the helix traced on the cylinder. In fig. 1 the pitch cylinder is represented by a dotted circle 0 3 E, which is the pitch circle of the wheel. The square 0 6 VI, H, fig. 6, is a development of the pitch surface, and the diagonal 0...VI that of the helix 0...VI. The line 0 C *z* is equal to one-half the circumference of the pitch circle, fig. 1, and the line H VI is one-half the pitch; each is therefore 6 inches. The angle of *inclination* VI, 0 H is 45°. The line H M is the normal

pitch of the *half coil* of the helix, and is drawn from H perpendicular to 0 VI, H p is the divided circular pitch $= \frac{\text{circumference of circle } 0\,3\,\text{E}}{\text{No. of teeth}} = \frac{12}{12} = 1$ inch.

H r is the divided axial pitch $= \frac{\text{pitch of helix}}{\text{No. of teeth}} = \frac{12}{12} = 1$ inch;

H t is the divided normal pitch $= \frac{\text{H M}}{\text{No. of teeth}} = \frac{\text{normal pitch}}{12}$

In this example we have only drawn out one-half of the development of the pitches, and so we must only take one-half of the quantities in our calculations; therefore $\frac{\text{F K and F L}}{\text{H } r \text{ and H } t}$ become $\frac{6}{6} = 1$ inch, and H $t = \frac{\text{H M}}{6}$.

Having drawn the centre lines for figs. 1 and 2, describe the pitch circle o 3 E, and draw the elevation o 6 VI D of the pitch cylinder, fig. 2. Then draw the helix 0...VI, as previously explained. Draw the development of the cylinder and of the helix, as in fig. 6, and the portion of the normal helix H M, which is one-half the normal pitch of the helix 0...VI; H M is drawn from H perpendicular to 0 VI. Divide H O into six equal parts H p, etc., and divide H VI into six equal parts H r, etc.; join p r, then p r will be parallel to O VI. If more convenient, divide only one of the lines and obtain p or r, from which draw p r parallel to O VI. The line p r cuts H M in t; H t is the divided normal pitch, while H p and H r are the divided circular and axial pitch respectively.

Now determine the radius of the circumference of the normal helix; in this case it is equal to the diameter of the pitch circle; however, the construction is shown in fig. 3, on the left hand of fig. 6. The angle of inclination of the helix being 45°, the radius of centre of the helix 0...VI and of the normal helix is the same for each, because both are inclined at the same angle; the construction shown is that required for the radius of the helix 0...VI. Upon the line O H mark off O T equal to the radius of the pitch circle of the wheel, and draw T U perpendicular to O VI to meet the developed helix 0 VI in U; from U draw U D perpendicular to O U, meeting O H in D. Then O D is the radius required. With

this radius describe an arc of a circle, as P Q, fig. 3, and treat this arc as if it were an arc of the pitch circle of an ordinary spur wheel, whose pitch is equal to the normal pitch of the screw wheel, viz., H *t*, fig. 6.

In fig. 5 we have shown a convenient method of obtaining the proportions of the teeth of wheels for different pitches; it is, in fact, a scale of proportions of the teeth of wheels from 0 to 1 inch pitch, and of course it may be extended to any dimension of pitch; its construction is as follows:—

Draw the straight line 0 A of any convenient length, from A draw A B perpendicular to 0 A, and equal in length to the greatest pitch the scale is intended to include; in the fig. A B is 1 inch long, for a pitch of 1 inch. From A, towards B, set off the proportions of the tooth from one of the sets previously given in Table XIII., page 193, and mark them as shown. From each of the points *c*, *d*, and *e*, thus obtained, draw lines to 0. Divide the line 0 A into such parts as will give the pitch required; in the figures, 0 A is divided into 4 equal parts, so that 3 perpendiculars at $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$, are respectively $\frac{1}{4}$ in., $\frac{1}{2}$ in., and $\frac{3}{4}$ in., and the perpendiculars are cut by the lines drawn from *c*, *d*, and *e*, which divide them in the same proportion as that line is divided.

Take the normal pitch H *t*, and draw H *t*, fig. 5, equal to it, so that it touches the lines 0 A and 0 B and is perpendicular to 0 A. Then H *t* is divided by the lines *c* 0, *d* 0, and *e* 0, in the required proportion for the different portions of the tooth. With these proportions describe the teeth as shown in fig. 3, making them according to one of the forms already described.

Draw the line yx at right angles to the radius S z, and upon it project lines from the top, pitch line, and bottom of the tooth Y. Now take the line yx, with the points 1, 2, and 3 upon it, and place it so that its middle part coincides with the point III of the helix, fig. 4; that is, where the projection of the helix and that of the axis intersect, and the line itself with the line $y'x'$, which is drawn at right angles to the direction of the helix at the point III. The line $y'x'$ approximates to the normal helix, and as only a small portion of it is used, it may practically be considered equal to it. From 1, 2, and 3, on $y'z'$, fig. 4, and perpendicular to it, draw lines to meet the line

$y\ z$ in 1, 2, and 3; these points give the width of the teeth on the top, pitch line, and bottom, for the elevation, fig. 1. Make the lengths of the tops and bottoms of the teeth in fig. 1 the same as those in fig. 4; that is, make 1, 2 in fig. 1 equal to 1, 2 in fig. 4. Describe circles through the points thus obtained, and upon them and the pitch circle set off the thickness from the line $y'x'$, fig. 4. The tooth Y′ is projected from the line $y'\ x'$, and the dotted circles correspond in diameter to those in fig. 1. We thus obtain in fig. 1 a short wide tooth, which of course we could anticipate, for the teeth are not perpendicular to the face of the wheel, fig. 1, but oblique to it.

The teeth in fig. 2 are projected in the same way as we project a screw, the width of the wheel is such that one tooth occupies as much of the pitch circle as a tooth and a half, as shown by the tooth E, fig. 1. With two such wheels in gear, just as one pair of teeth quit contact the next pair will be half and half in gear, so that there will never be less than the halves of a pair of teeth in contact.

328. Pair of Unequal Screw Wheels in Gear.—On Plate XXII. are shown a pair of unequal screw wheels in gear, the teeth or threads are right-handed, and their axes are at right angles. The smaller wheel D, fig. 16, is the driver, and the projection of its axes are marked $a\ a$, $a'a'$; the larger wheel E is the follower, and the projection of its axis are marked $b\ b$, b'. The wheel D has 6 teeth, and the diameter of its pitch circle is 2 inches; the wheel E has 18 teeth, and the diameter of its pitch circle is 2 inches; therefore the velocity-ratio of the axes A and B, A being the axis of D, is $\frac{A}{B} = \frac{18}{6} = 3$; that is, A makes three revolutions to one of B, while the ratio of the diameters of the wheels D and E is $\frac{2}{4}$ or $\frac{1}{2}$.

These wheels have already been shown in plan and elevation, and in figs. 293, 294, we again show them in plan, fig. 295, to which figure we now refer. The construction lines shown on this figure are repeated on Plate XXII., but for clearness we refer to this figure to describe the first part of the construction.

Draw the projections $a\ a$ and $b\ b$ of the axes A and B, and let them intersect in the point C. From C, on the line $b\ b$,

and towards b, mark off any convenient length $C f$ proportional to the angular velocity of rotation of the wheel E; and upon $a\,a$ mark off $C g$ proportional to the angular velocity of rotation of the wheel D; that is, $C f$ in $C b$ is one unit in length, and $C g$ in $C a$ is three of the same unit in length; join $f g$ and divide it in h, so that

$$f h : h g :: \text{radius of D} : \text{radius of E};$$

that is, inversely proportional to the radii of the wheels D and E.

Through C and h draw the line F C F, and through C, at right angles to F C F, draw G C G.

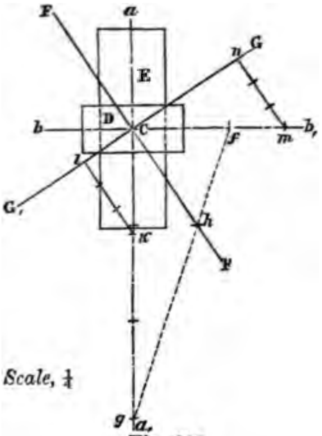


Fig. 295.

In the case of screws, and therefore of screw wheels, it is important to notice that, for a *right-handed* screw, the thread on the back half of the cylinder appears *left-handed*, and *vice versa*. In the case of two right-handed or two left-handed screw wheels in gear, it will depend upon which of the two wheels we make the upper one, whether the line of contact G C G is inclined, as shown in fig. 295, or not. In this figure it is inclined in the wrong direction if we make the smaller wheel D the upper one, in which case the point f should be on the opposite side of C towards b. But if we follow out this system it would entail more explanation, and, we think, lead to confusion; therefore we shall simply give the one construction, as shown in the figure, for obtaining the lines F C F and G C G, and the rest will be clear from what is to follow.

Having drawn the line G C G, the angle G C f is the angle *of indication* of the threads of the screw of the wheel E, and the angle G C g that of the wheel D. If the wheels are right-handed, then draw the developments of the threads

accordingly, and with the given inclinations as in the example to follow. The wheel D has 6 teeth and E has 18, and the diameters of their pitch circles 2 inches and 4 inches respectively; therefore the length of the circumference of D's pitch circle is 2π inches, and of E's 4π inches.

In fig. 295 we have made $Cm = 3{\cdot}1416$ inches $= \frac{1}{2}$ the circumference of D's pitch circle, so that Cm contains 3 teeth, and $Ck = \frac{4 \times 3{\cdot}1416}{6} = \frac{1}{6}$ the circumference of E's pitch circle, so that Ck contains 3 teeth. From either k or m draw kl or mn perpendicular to G C G, and meeting it in l or m; divide either kl or mn into three equal parts, then one of these parts is the length of the *divided normal pitch. The divided circular and axial pitches* we will refer to in the following example.

329. In figs. 16-29, Plate XXII., are shown the pair of unequal screw wheels, partly described in the previous articles, drawn to a scale of one-half. Fig. 16 is a plan of the pitch surfaces of the wheels D and E, with the necessary construction lines, as described in the previous article. Fig. 17 is a portion of the cylindrical pitch surface, larger than necessary, of the wheel E, showing $\frac{1}{6}$ of the helix; fig. 18 is an elevation of a portion of fig. 17; fig. 19 is a development of this portion of the helix. Fig. 20 is a portion of the pitch surface of the wheel D, showing $\frac{1}{2}$ of the helix; fig. 21 is an elevation of fig. 20; fig. 22 is a development of 20; fig. 25 is a section of the pitch surfaces; figs. 26-29 are plans and elevations of the two wheels.

Draw the projections $a\,a_1$, $b\,b_1$, $a'b'$, fig. 16, of the axes A and B intersecting in C, and determine the points f, g, and h, and draw the line F C F, G C G, as just described. Draw also the cylindrical pitch surfaces of the wheels D and E, at present we need not give any particular width for these. Now draw the portion of the cylindrical pitch surface of E, fig. 17, and on the left of fig. 16 draw a circle, or a portion of one, 0 6 0, fig. 18, having b' for its centre, and of the radius of the pitch circle of E (2 inches). From 0 on the circle, fig. 18, mark off 0...3, equal to $\frac{1}{6}$ of the circumference of the pitch circle, and divide the arc 0...3 into any convenient

number of equal parts, say 3. The arc 0...3 is $\frac{1}{6}$ of the circumference of the pitch circle or circular pitch of the wheel E, and therefore contains 3 teeth. Mark *o* H, fig. 19, equal to 0...3, fig. 18, and draw H III parallel to *o* K; make the angle K *o* III, equal to the angle of inclination of the helix on E, equal to the angle G C *f*, fig. 16. From H draw H III parallel to *o* K to meet *o* III in III; from III draw III K and produce it to L, and complete the outline of fig. 17. The line *o* K is $\frac{1}{6}$ of the axial pitch, and *o* H $\frac{1}{6}$ of the circular pitch of the helix. From H draw H M perpendicular to *o* III, then H M is $\frac{1}{6}$ of the normal pitch of the helix. Divide either *o* H or H III into three equal parts in *q* and *p*, draw *p r* and *q s* parallel to *o* III; these lines divide H M into three equal parts. Then the lines H *p*, H *r*, and H *t* are respectively the divided circular, axial, and normal pitches of the helix, of which *o* III in fig. 17 is $\frac{1}{6}$.

Draw figs. 20, 21, and 22, which are two projections of a development of a portion of the cylindrical pitch surface out of which the wheel D is formed. The wheel D has 6 teeth, therefore in each half circumference there are 3 teeth; the the line *o* H′, fig. 22, is equal to half the circumference of the pitch circle of D, and the angle K′ 0 VI, marked θ, is equal to the angle of inclination of the helix on D. The figs. 20 and 22 are drawn in a similar manner to figs. 17 and 19; the letters of reference are the same with accents, we need not therefore repeat the construction. The lines H′*p*′, H′*r*′, and H′*t*′, are respectively the divided angular, axial, and normal pitch of the helix, of which 0...VI, fig. 20, is one-half. The divided normal pitches H *t* and H′*t*′ are equal; this must be the case in all such wheels. The divided circular pitch H *p* of the wheel E is equal to the divided axial pitch H′*r*′ of the wheel D; and the divided axial pitch H *r* of the wheel E is equal to the divided circular pitch H′*p*′ of the wheel D. The equalities just named hold in all cases of the same wheels with axes at right angles.

The diameter of the pitch circle of D is 2 inches, and it has 6 teeth; therefore its divided circular pitch

$$=\frac{2\pi}{6}=\frac{2\times3{\cdot}1416}{6}=1{\cdot}0472 \text{ inch.}$$

The diameter of the pitch circle of E is 4 inches, and it has 18 teeth; therefore its divided circular pitch

$$=\frac{4\pi}{18}=\frac{4\times 3{\cdot}1416}{18}={\cdot}698 \text{ inch.}$$

The divided axial pitch of D and E are respectively ·698 and 1·0472 inch. Therefore the total axial pitch of the helix for $D = 1{\cdot}0472 \times 6 = 6{\cdot}2832$ inches; and that of the helix for $E = {\cdot}698 \times 18 = 12{\cdot}564$ inches.

In fig. 23 are shown two developments, figs. 19 and 22 combined. The width of the wheels may be obtained from this figure; let us say the width of each shall be such that, in figs. 27 and 29, the projections of a tooth shall occupy a pitch and a quarter, measured from centre to centre of tooth along the pitch circle. That is to say, just as a pair of teeth quit contact, the next pair will be in contact to the extent of $\frac{1}{4}$ of their length of contact. We may obtain the necessary width for a given length of contact by means of fig. 23. The line o H represents the circular pitch of the teeth of the wheel E, and the line o H′ that of the teeth of the wheel D. $o\,q$ and $o\,q'$ are respectively $\frac{1}{3}$ of o H and o H′. Upon o H or o H′ make $o\,v$ equal in length to the divided circular pitch, and the fraction of that pitch of the wheel E or D for the desired length of contact in the figure $o\,v = 1\frac{1}{4}$, the divided circular pitch. From v draw $v\,w$ parallel to o H′, and $w\,v$ parallel to o H. Then $o\,v$ in o H will be the width of the wheel D, and $o\,v$ in o H′ that of the wheel E.

In fig. 25 is shown a quarter of the section of each cylinder made by the plane whose horizontal trace is F C F; this plane is normal to the direction of the two helices at the point C, fig. 16. The section of D is marked 0...III, and of E, 0...VI; the construction of these sections need not be given here, as they have been fully explained already. We have drawn the sections to show how near the circles N C′0, P C′Q, whose radii R C′, S C′ are the radii of curvature of the section at the point C′, approximate for a short distance on each side of the point C′, to the portion 0 1, 0 1 of the quarter ellipse 0...III, 0...VI.

The construction for obtaining the radius of curvature R C′, S C′ is shown in fig. 16, and also in figs. 19 and 22; in

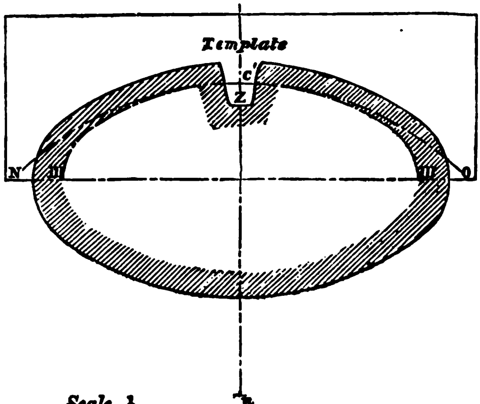
the latter, the letter *o* is substituted for C′, and in fig. 16, C for C′. We shall refer to fig. 16. Make C T in the axis *bb*, the radius of the pitch circle of the wheel D; from T draw T U perpendicular to C T, meeting F C F in U; and from U draw U R perpendicular to C U, meeting C T produced in R; then R C is the radius of curvature of the normal helix of the wheel D, and with this radius the pitch circle N C′O is described. To find S C′ make C W in C *a* equal to the radius of the pitch circle of the wheel E; draw W 6 perpendicular to C W, meeting F C F in 6; and from 6 draw 6 S perpendicular to C 6, meeting C W produced in S. Then S C is the radius of curvature for the wheel E. With these radii describe arcs of circles N C′O and P C′Q, and set off upon them the normal pitch H *t*, fig. 19, of each wheel. Now describe a tooth and space on each of these pitch circles for wheels of a pitch H *t*, as shown at X and Y, fig. 25, and of the desired form of tooth. In fig. 5, Plate XXI., the line H′*t*′ is equal to the normal pitch of these wheels; that is, H′*t*′, fig. 5, = H *t* and H′*t*′ on Plate XXII.

330. The projections of the wheels, figs. 26-29, are obtained in a similar manner to that shown in Plate XXI., and described when those figures were explained; and as the same letters of reference are employed, we refer the student to that example, page 287. The dotted lines, O...III, O...VI, figs. 26 and 28, are portions of the developed helices. In each of these examples we have determined the form of tooth, excepting as regards the particular curve given to the acting surfaces, and which are already described by its *normal section*, just as the tooth of a spur wheel is determined.

Rankine, in *Machinery and Millwork*, first edition, 1869, states that "this method of describing the threads of screw-gearing is believed to be now published for the first time." This may be so, but we must say we saw, some 12 or 14 years ago, wheels which were, we believe, constructed upon this very principle, although we do not think any rules of a general kind were laid down or worked upon in this case; and, in addition, we arrived at the very same conclusion, that the correct way of describing such teeth was by means of the normal section, before we had read Rankine's construction or seen his statements. As these wheels have

been in use for many years, although only used occasionally, we think it advisable to make the foregoing statement.

331. In fig. 296 is shown a *template*, which may be used either when turning the wooden pattern or the metal wheel.



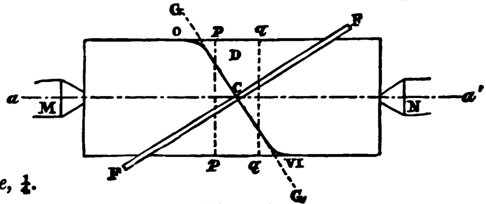
Scale, ½

Fig. 296.

The fig. represents a section made by the normal plane, F C F, of a cylinder, out of which the smaller wheel D is to be made, Plate XXII. The position of this section plane is shown there, as is also the line GCG, and half a turn of the helix, marked 0...C...VI. The dotted semi-ellipse III...C′...III, fig. 296, represents one-half the section of the pitch surface, while the larger ellipse represents the section of the whole cylinder; the section III...C′...III corresponds to the section shown in fig. 295. The point R is the centre of curvature, and the arc N C′ O is an arc of the circle described with the radius of curvature R C′. The tooth-like piece Z is of the same size as, and corresponds to the space Z, fig. 25, between two teeth; this, of course, is the section of one of the grooves that are to be cut, of which six are to be cut in the cylinder, the projecting threads or teeth left being the teeth of the wheel.

To use this template it is simply necessary to draw a line F C F, fig. 297, upon the cylinder, at right angles to the direction of the threads that have been traced or cut slightly

upon the cylinder; in the figure the position of the template is represented by two parallel lines, F C F. The curve 0 C VI is one-half a turn of the helix, and the line G C G is a portion of the developed helix, and it corresponds to the line G C G, fig. 295; the dotted lines *p p* and *q q* represent the width of the wheel D. The line *a a* is the centre line of the cylinder, and also represents the axis of the lathe, of which M and N are its centres. The template may have



Scale, ½.

Fig. 297.

the piece Z movable, so that it may be replaced by pieces of varying size, increasing from a small piece to the full form shown in the figure, or two or more templates, with similar increasing sizes of the piece Z, may be employed. As the wooden patterns and the metal wheels, if they are cut and not cast, would be cut in a screw-cutting lathe, the helix and the normal helix may both be traced upon the cylinder which is to form the wheel by means of such a lathe having the necessary change wheels.

332. We have already stated briefly the peculiarities of the worm and worm wheel; we now propose to consider the form of the teeth, and to give examples. In figs. 7 and 8, Plate XXI., is shown a worm and worm-wheel in gear. The worm D has a single thread, and the wheel E has 30 teeth; the projections of their axes A and B are *a a, a'*, and *b, b' b'*. It was shown by Willis that if through the axis of the worm, perpendicular to the axis of the wheel, we draw a plane, and upon that plane the teeth of the worm and worm wheel are described as if they were for a rack and wheel, then the worm and worm wheel will gear correctly. In fig. 8 this plane is shown by its trace α β, and in fig. 7 the portion of the worm and worm wheel in section is made by that plane.

In speaking of the form of teeth we shall always refer to the form as described upon the central plane $\alpha\,\beta$, fig. 8, and represented by the portion of the worm and worm wheel in section in fig. 7.

The form given to the teeth may be any of those previously described for the rack and pinion. In figs. 7 and 8 the thread of the worm is rectangular, and the teeth of the wheel have involutes of their pitch circles for their acting surfaces. The outer diameter of the worm, as seen in fig. 7, is a tangent to the pitch circle of the worm wheel. The teeth, still keeping the straight line form for the worm and involutes for the wheel, may be similar to those of the rack and pinion. In figs. 7 and 8 the tops and bottoms of the teeth and the pitch surface are cylindrical; that is, the lines that represent these in fig. 8 are parallel to the axis. The teeth are in fact similar to the teeth of a spur wheel, but instead of being parallel to the axis they are inclined to it at the *angle of inclination* of the helix. The lines forming the tops and bottoms of the teeth, as *f f*, *g g*, *h h*, fig. 8, are straight lines, and are portions of the developed helical lines; these portions being relatively very small, they approximate nearly to the true helical form.

In figs. 298 and 299 is shown another form of tooth for a worm and worm wheel. In this case the form of the teeth, as described on the central plane, whose trace is $\alpha\,\beta$, may be any of the forms named for those of a rack and pinion. The teeth, as seen in fig. 299, however, are not formed by straight lines; the traces of the tops and bottoms of the teeth are arcs of circles described from a' as a centre, and the pitch line coincides with the pitch circle of the worm.

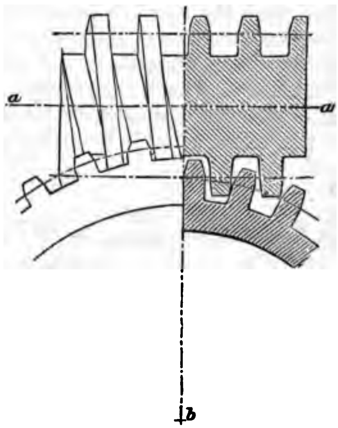


Fig. 298.

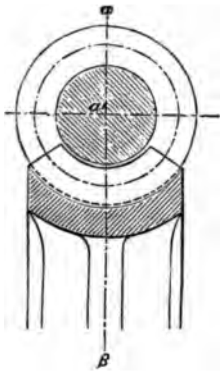


Fig. 299.

In fig. 298 the right-hand half is shown in section, the section being made by the central plane $\alpha\,\beta$, fig. 299; the left-hand half is a projection of the outer surface. The teeth, as described on the central plane $\alpha\,\beta$, are similar to those of the rack and pinion, Plate XII. The teeth of the worm are formed of cycloids for the tops, and straight lines for the bottoms; and those of the wheel, of involutes of the pitch circle for the tops, and radial lines for the bottoms.

333. Worm and Worm Wheels in Gear.—In figs. 7, 8, and 9, Plate XXI., is shown a worm and worm wheel in gear; the worm D has a single thread and is right-handed; the worm wheel E has 30 teeth and is right-handed. The pitch is 1 inch; that is, the axial pitch *r s*, fig. 7, of the worm is 1 inch, and the circular pitch *p q* of the wheel is 1 inch. In the case of a worm and worm wheel we use the term pitch in the same sense as in the case of ordinary spur wheels or a rack and wheel. The pitch circle of the worm coincides with its outer diameter, which is 3 inches. The pitch circle of the worm wheel is found by the usual formula—

$$D = \frac{P}{\pi} \cdot N.$$

In this case $D = \frac{P \times 1}{3 \cdot 1416} \times 30 = 9 \cdot 549$; say $9 \cdot 55$ inches. The teeth of the worm wheel have involutes of their pitch circle for their acting surfaces, while the thread of the worm is of a rectangular cross section, as seen on the right-handed half of fig. 8.

Having drawn the centre line, describe the pitch circle F C G of the wheel, and set out the centres of the teeth as shown. On the right-hand side, the three teeth K, L, M next to the centre line H C *b* are in section, so that in setting out the centres of the teeth allowance must be made for this circumstance. From C set off, upon the pitch circle, the

thickness of a tooth from one of the sets of proportions given in Art. 236, or from fig. 5, Plate XXI.; find the centre of the tooth, and set out the centres of the two teeth L and M.

From C describe a portion C *c* of the involute of the pitch circle F C G, as shown already by the construction lines. The acting surfaces of the teeth are found by the portion of the involute C *d*. Inside the pitch circle draw a circle for the bottoms of the teeth, allowing clearance only, as the whole acting surfaces of the teeth are formed of portions of the involute; the lines between these circles are radial lines, as shown at C. Also describe a circle for the tops of the teeth—in the present case the length of the tooth outside the pitch circle is equal to the thickness of a tooth, and draw the portions in section of the teeth L and M.

From C mark off along C H the depth of the thread of the worm, which is equal to the length of a tooth of the wheel. Draw the end elevation, fig. 8, of the worm, and draw the outline of the worm, fig. 7. From C draw the curve C *k*, the construction of which is shown in fig. 9; 0 *m*, fig. 9, is equal to half the pitch, and the curve 0...VIII is a projection of one-half the helix. Having drawn the curve C *k* mark off from C, along N C O towards O, a distance equal to the space which is the same as for the wheel, and so divide the whole length of the worm into spaces and projections, which represent the thread. The curves for the tops and bottoms of the thread are drawn as already described.

On each side of the centre line H C *b* two teeth of the wheel are shown in contact with the thread of the worm, contact taking place in the central plane, as shown by the portion in section and by the dotted lines on the left-hand of the centre line. The worm and wheel are both assumed as turning round right-handed, as shown by the arrows, the worm driving the wheel. Having now drawn the worm and the mid-section of the wheel, we have to determine the inclination of the teeth of the wheel, and so complete the two figures of the wheel. The projections *f f*, *g g*, etc., of the teeth of the wheel, as seen in fig. 8, are straight lines, which are the portions of the developed helices of the worm wheel. The helices may be developed in a similar manner to those for screw wheels; but in the present case, where the angles of

inclination are at the two extremes, it is simpler to proceed as follows :—

Make a triangle 0 P 8, fig. 9, one side P 8 being equal to the semi-circumference of the pitch circle of the worm, which in this case coincides with the outer circumference of the worm, while the short side 0 8 is equal to one-half the pitch. The angle 0 P 8 is the angle of inclination of the teeth of the wheel; in fig. 9 the development of one-fourth of the pitch circle of the worm is shown in the lower portions of the figure, together with the constructions for obtaining it; the line 4...P is equal to the arc 4...8. In fig. 9 the curve 0...VIII is the projection of one-half of a turn of the helix forming the thread of the worm, and the straight line 0 P is the development of this curve, a portion of which is employed for the projection of the teeth of the wheel, and it will be seen that as the breadth of the wheel is less than one-half of the line 8...P, the curve and its development nearly coincide. In fig. 7, T U is equal to the width of the wheel, and the angle U T V is equal to the angle 8 P 0, fig. 9. Upon the pitch circle make $u\ v$ equal to U V, and draw $v\ v'$ parallel to $u\ u'$; make $v\ x$ equal to $u\ w$, and draw $x\ x'$ parallel to $w\ w'$, then the tooth Q is complete. The dotted line $x\ x'$ is not shown on the other teeth, but in making an accurate projection it is necessary to draw this line upon all the teeth projected. The distance $u\ v$ may now be set off for each tooth, except for the three in section, where only one-half of this distance is used, as the section plane passes through the middle of the wheel. In drawing the projections of the teeth in fig. 8, lines are drawn from w' and u' upon one edge of the wheel, and from x' and v' upon the other edge, then w' and u' and x' and v' are joined, forming the projection of the top of the tooth. The bottoms of the tooth may be projected in a similar manner. It is usual to join the teeth to the rim with small curves, in which case there would be no lines to represent the bottoms in fig. 8; but some of the teeth, those near the horizontal centre line passing through b, will have lines to represent the width on the pitch circle, as is the case in the projection of a spur wheel. For example, if we project a plane of fig. 7, then the tooth K would have two lines for the top of the tooth and two for the thickness on the pitch circle.

CHAPTER IX.

Eccentric—Machine Tools—Steam Engines and Boilers—Steam Hammer—Pair of Double-Action Pumps.

SECTION I.

THE ECCENTRIC.

334. THE eccentric is a well-known means of converting circular motion into rectilinear reciprocating, but it can only be employed where the reciprocating motion required is very limited. The eccentric possesses the great advantage that it can be attached to the shaft without necessarily being fixed at one end, and without causing a break, as the crank does, in the length of the shaft. Plate XXIII. shows an eccentric. It consists essentially of a circular plate A, sometimes termed a sheave, which is keyed to the shaft S. The principle upon which it acts has been already fully illustrated; it now remains to explain the construction of this piece of machinery more fully. The centre of the sheave and that of the shaft are a certain distance B C apart, this is termed the *eccentricity;* twice the eccentricity B C (= B D) is termed the *throw.* As the sheave is fixed to the shaft it turns with it; the motion is taken from the sheave by means of the *strap* E, which consists of a ring in halves fitting into a groove cut in the sheave and connected by bolts F, F. The strap does not turn round with the sheave, but oscillates, having B for a centre; and at the same time it receives a motion in directions B D, D B; therefore the strap must not fit the sheave too tightly. At G are inserted pieces of metal or hard wood, by adjusting the thickness of which, compensation can be made for the wear between the surfaces of the sheave and

the strap. Attached to the strap by means of bolts, or by a *cotter*, as in the figures, is a rod H K, which transmits the motion to the piece to be operated upon, as the slide-valve of the steam engine in the example, where L is one end of the valve-rod.

335. The dimensions of the several parts of the figures shown in Plate XXIII. are as follow:—The throw B D is $2\frac{1}{2}''$, the diameter of the shaft S is $3\frac{1}{2}''$, the key T is $\frac{3}{4}'' \times \frac{7}{16}''$, and it is let into the shaft $\frac{1}{8}''$. The sheave A is $7\frac{3}{4}''$ diameter outside, $7\frac{1}{4}''$ diameter at the bottom of the groove, and $2''$ wide; the width of the groove is $1\frac{1}{2}''$; the thickness of metal round the shaft is $\frac{13}{16}''$, at g it is $1''$ thick to allow for the key-way; the rim e is $\frac{1}{2}''$ thick, and the arm f is $\frac{3}{4}''$ wide. The strap is $7\frac{1}{4}''$ diameter inside, and $8\frac{1}{4}''$ diameter outside; the width is $1\frac{1}{2}''$ *bare;* each half is provided with *lugs* $\frac{5}{8}''$ thick, through which pass bolts F $\frac{3}{4}''$ diameter; the distance, centre to centre, of the bolts, is $9\frac{1}{2}''$; at V, on one half of the strap, is a boss $1\frac{1}{2}''$ diameter, to receive one end of the rod H K; U, U, are feathers $\frac{1}{2}''$ wide, whose object is to strengthen the connection between the boss and the strap; W is a collar on the boss V $1\frac{5}{8}''$ diameter and $\frac{3}{8}''$ wide; the distance from the outside of the collar to the centre B of the strap is $6''$. The cotter Q is $3''$ long, $\frac{3}{16}''$ thick, and $\frac{7}{8}''$ wide in the middle; the amount of *taper* in its length is $\frac{1}{2}''$ per foot; M is an oil-cup forming part of the strap, a section of which is given in fig. 5; R is a hole through which the oil passes; the cup is $1\frac{1}{2}''$ diameter outside, and $1\frac{1}{4}''$ diameter inside; the tube is $\frac{1}{2}''$ diameter, the hole $\frac{1}{4}''$ diameter; the distance from the top of the cup to the centre line is $4\frac{1}{8}''$; the cup is provided with a cover O, which is screwed into the cup; the diameter of the screwed part is $1''$, but the thread is finer than that usually given for one inch diameter. The edge of the cup-cover is generally *milled*, to allow of a better hold being taken when unscrewing it. The other dimensions may be taken from the figures. The eccentric rod H K is $2'...5''$ from the centre P to the outside of the collar; the portion in the boss V is $1\frac{7}{8}''$ long and $\frac{7}{8}''$ diameter; the rod is $\frac{7}{8}''$ diameter at each end, and increases to $1\frac{1}{8}''$ in the middle; the end K of the rod is *forked*, and through it passes a pin X, connecting the valve-rod L to it;

between the fork and the cylindrical portion K the cross-sections are rectangular and square; a portion of the latter has its edges chamfered, leaving the section an octagon. The pin X is prevented from leaving its position by means of a pin which passes through the former; between the pin and the fork is a washer 1″ diameter and $\frac{1}{8}''$ thick; the pin may be either a piece of round *wire*, or in the form of a *split-pin;* the cross-section of the wire out of which it is made is a segment of a circle, nearly a semicircle; by opening out the halves of the pin at *a* it is prevented from leaving the hole in which it is placed. The sheave is cast-iron, the strap is brass, and the rod, pin, washer, and cotter, are wrought-iron.

336. Fig. 1, Plate XXIII., represents in outline the eccentric arrangement; the centre line $a' y'$ is the path of the valve-rod, which passes through the centre C of the shaft; B E D F is the path of the centre of the eccentric; B D is the throw; the positions *b*, *d*, of the rod end correspond to the positions B, D of the eccentric; $b d = \mathrm{B\,D}$. The sheave is shown in four positions, I, II, III, IV, whose centres are B, E, D, F, respectively; the variable motion obtained from this arrangement is similar to that obtained from the crank.

337. The general problem is, given the throw of the eccentric and the diameter of the shaft upon which it is to be fixed, to make a drawing of the arrangement. In Plate XXIII., figs. 2 to 4, we have worked out the example, of which the dimensions are given in Art. 335, p. 300. Fig. 2 is a front elevation, fig. 3 is a plan, and fig. 4 is an end elevation; they are drawn to a scale of $\frac{1}{4}$. Fig. 5 shows a portion of the sheave and the strap, with the oil-cup in section.

338. The drawing of the figures in Plate XXIII. is as follows:—Draw the centre lines $a y$, $a' y'$, $b z$, $c x$; let C be the centre of the shaft; from C as a centre with a radius C B (one-half the throw), describe a circle B D N, which is the path of the centre of the sheave; through B draw the line $d w$, this will be the centre line of the sheave and the strap. From C and B as centres describe the circles for the shaft, etc.; and from the dimensions given, and from the construc-

tion lines shown, proceed to draw the figures. The only special points to be noticed are the intersections of the oil-cup with the strap, of the feathers U with the boss V and with the strap, and of the boss with the strap, which are fully worked out in the Elementary Volume of this series.

SECTION II.

MACHINE TOOLS.

Slot Drilling and Grooving Machine—Self-Acting Slide and Screw Cutting Lathe—Horizontal Boring and Surfacing Machine.

339. The term machine tool denotes a machine used for the purpose of performing work upon some object on which it is intended to make a change of form. This change of form may be brought about by cutting, forging, grinding, etc. We have thus many operations and an infinite variety of objects to be operated upon. The very great improvement that has been made in mechanical engineering during the past thirty or forty years could not have been effected, if during that time and just preceding it, there had not been very great improvements made in machine tools.

It is within the memory of some of our elder engineers that the only tools they possessed were the lathe and the drill, and these were very primitive, indeed, compared with those of the present day. The first great improvement of the lathe was the slide rest, which has been introduced in various forms into many modern tools. In fact the principle of the slide is the basis of all our improvements. After the slide rest, planing machines were introduced, and then followed slide lathes with compound slide rests, etc. These three tools, the lathe, the drill, and planing machine, have been followed by an almost indefinite number, which is ever increasing, and will continue to do so, as long as special tools are required; for it is the developement of the labour-saving tools that has so materially increased the ordinary tools of ten and twenty years ago. It may be safely

said that no engineering work of the present day need be abandoned for want of the appliances which it is the special work of the tool maker to supply.

In this section it is proposed to give some examples of modern machinery, illustrating the more elementary examples that have been explained in the preceding pages. For this purpose machines have been selected that are readily divisible into three classes. It would be perfectly impossible to illustrate every class of machines, therefore those are selected that seem to embrace fundamental principles, and which are in general use, and important to every mechanical engineer, and to every student of machine mechanism. In the first class machine tools are illustrated; in the second, steam engines and boilers; and in the third a steam hammer; after which are added a few remarks on a double-action steam pump.

340. Slot, Drilling, and Grooving Machine.—Plates XLII., XLIII., XLIV. In figs. 1-6, Plate XLII., is shown in elevation and plans a patent self-acting slot, drilling, and grooving machine, manufactured by Messrs. Sharp, Stewart, & Co., Manchester. It is single geared and for general work. As its title implies, this machine is for a special class of general work, such as cutting key beds in shafts, cranked axles, etc.; slots for cotters in connecting rod ends; in the strap for the same; in piston-rods, cross-heads, etc., and for a variety of similar work. The drilled holes are generally of a longitudinal section, and are cut at one operation.

Fig. 1 is a front or end elevation; fig. 2 a longitudinal elevation; fig. 3 is a longitudinal and sectional elevation, made by various planes, but chiefly by a vertical plane passing through the centre line $\beta\,\beta$, fig. 1, and $\eta\,\eta$, fig. 4. Fig. 4 is a sectional plan made by a horizontal plane passing through $\delta\,\delta$, fig. 1; fig. 5 is a plan showing the positions of the elliptical wheel, etc., which moves the handstock; fig. 6 is a plan, partly in section, showing the gearing for moving the headstock.

Plate XLIII. In this Plate is shown an enlarged end elevation, partly in section, the chief portions of the section is made by a vertical plane passing through $\delta\,\delta$, fig. 4.

Plate XLIV. In the two figures on this Plate are further details, showing principally the self-acting feed motion. Fig. 9 is a front (partial) elevation, with portions in section, showing the right-hand end of fig. 2 and the headstock; and fig. 8 is an end elevation, with portions removed, to show more clearly the part previously mentioned. The machine consists of a bed supported in standards, a headstock which moves upon this bed, and which carries the drill spindle and the gearing for driving it, and a table upon which is fixed the shaft or other piece of material which is to be operated upon. The work under manipulation is stationary, and the drill moves with the headstock, which has a range or traverse of 21 inches. Slots of from 0 to 12 inches long by 9 inches deep and 3 inches wide, can be drilled by this machine with one operation. More than this can be done, to which allusion will presently be made.

341. For the purpose of describing the several parts, two divisions are made, as follows:—

DIVISION I. The *Bed, Standards,* and *Gearing* for moving the Headstock.

DIVISION II. The *Headstock* and *Self-acting Feed Motion.*

I. The Bed, Gearing, etc.—The student must clearly understand that the same letters will be found in all three Plates XLII., XLIII., XLIV., and that they indicate, wherever found, the same piece of mechanism. Where the same letter is employed twice on a drawing, it in one case refers to Division I., and in the other to Division II. The bed A is a cast-iron box-frame, of the form shown in the several figures; at each end, on the top side of the bed, is a bracket A cast to it (seen most clearly at the upper A in fig. 1), these brackets form the bearings for the driving shaft. The bed rests upon two standards B B, which are connected to it by bolts and nuts *a*...*a* (fig. 3). The driving shaft C is carried by bearings *b b*, this shaft has upon it ten cone pulleys D and E. The former is made up of three speeds D, D_1, D_2, and connected by a strap with the *counter* shaft from which the machine receives its motion; a section of this pulley, exhibiting the mode of attachment to the shaft, is shown in figs. 7

and 8. The shaft is maintained in position in its bearings by the boss of the cone pulley D, and by a loose collar *c* (figs. 2 and 3), which is fastened to the shaft by a pin. The pulley E is also keyed to the shaft, and it bears against the other bearing *b*. The pulleys D and E are further connected to the shaft C by washers and set screws $c_1 c_1$. The shaft C carries another cone pulley of ten speeds connected with the feed motion, the pulley has fixed in it a sliding key as shown in fig. 3, while the shaft has a key-way cut in it to receive the key. The pulley moves along the shaft C according to the position of the headstock. Besides the pulleys mentioned, there is upon this shaft a bevel wheel (fig. 4) which transmits the motion from the driving shaft to an intermediate shaft, and hence to the drill spindle. The cone pulley E has six speeds $E, E_1, E_2, \ldots E_5$, the motion from the driving shaft C is transmitted from this pulley by a belt to the cone pulley F of six speeds $F, F_1 \ldots F_5$, which is keyed on the shaft G; this shaft is the driving or first motion shaft for actuating the headstock, it is supported in bearings $d\, d_1$ (fig. 3), cast on the standards B B. The pulley F is keyed to the shaft G, as shown in section, fig. 3, at F. At the other end of G is a collar *e*, and a loose collar e_1 on the inner side of the bearing d_1; the collars keep the shaft in position. The cone pulley F has its six speeds so arranged that the same strap may be employed to connect any one of its speeds with the corresponding speed on the pulley E; that is E and F; E_1 and F_1, etc., are pairs, the sum of whose diameters are equal. Upon the shaft G is fixed a right-handed worm H, working in a worm wheel K, which is keyed on a vertical shaft L (see fig. 3, Plate XLII. and Plate XLIII.); this shaft is carried in bearings *f* and f_1, the former projecting below the bottom of the bed, while the latter is in the bar *g* which passes from the front to the back of the bed, and is in one casting with it, the bar carries and acts as the bearing for the next shaft in order. Between the bearings *f* and f_1, and keyed to the shaft L, is a spur pinion M (figs. 3 and 7) in gear with a spur wheel M_1, which is keyed on the shaft N. This shaft is supported in a bearing *h* in the bar *g* of the bed, and upon its upper end is keyed a non-circular eccentric wheel O; in the figure this wheel is shown as a circular

eccentric one, which it very nearly approaches. The eccentric wheel O is in gear with an elliptical wheel P which forms its own shaft, and is carried in a bearing k in the bed of the machine; it is supported in this bearing by the cap k_1, which is held in position by the bolts and nuts l and l (fig. 3 and 6). The elliptical wheel P has upon its upper surface a disc or plate m, across this disc there is a T-headed slot m_1, on each side of which the metal of the disc is raised, fig. 5. The slot m_1 has its centre line in the same vertical plane that contains the minor axis of the ellipse, which forms the pitch line of the elliptical wheel P in all various figures; this centre line is at right angles to the centre line $\gamma\,\gamma$ of the machine, and a vertical plane containing the centre line passes through the centre of the elliptical wheel and the wheel O which actuates it. In fig. 6 the wheels O and P are shown in their two extreme positions, the elliptical wheel with its major and minor axis in the centre line $\gamma\,\gamma$, and the non-circular wheel O with its least and greatest radius in the same line. The non-circular eccentric wheel O is made up of two equal and similar halves, and the circumference of each is equal to the circumference of one-fourth of that of the elliptical wheel P; hence P must have twice as many teeth as O. They are so connected that the least radius of the wheel O is in contact along the centre line $\gamma\,\gamma$ with the greatest radius of the elliptical wheel P. The pitch line of the wheel P is a true ellipse, and it is centred at its geometrical centre, that is, at the point of intersection of its major and minor axis; the wheel O is, as already stated, a non-circular eccentric one, and is of such a form, that for every position of the two wheels when in gear, the sum of the radii is equal to the distance between the centres. But as the ellipse which forms the pitch line of wheel P and all its radii is fixed, the radius for any position of the wheel O is equal to the distance between the centres of the wheels O and P minus the corresponding radius of the wheel P; a radius of the latter being a line drawn from any point in the circumference to the geometrical centre of the ellipse.

342. The motion for actuating the headstock is transmitted from the elliptical wheel by means of a connecting

rod Q (figs. 3, 5, and 6), in one end of which q is fixed a bush n, through which passes a bolt o. This bolt has a head upon it which fits in the T-headed slot m_1 in the disc m, and by means of the nut upon it the bolt can be fixed in any required position in the slot, and with the bolt move the end q of the connecting rod Q. The other end q_1 of the rod Q carries a connecting pin R, through which passes the shaft S, the pin R is attached to the end q_1 of the connecting rod Q, with a washer and set screw. The shaft S is carried in bearings p and p_1, formed in the bed of the machine, the bearing p_1 is formed by a bush let into the frame. The shaft is free to move lengthwise and to turn, but both motions are governed by other movements which will be indicated presently. One end of the shaft S is squared to receive a handle for turning it; between the bearing p_1 and the piece R the shaft is of a larger diameter than the other parts, and has a right-handed square-threaded screw S_1 cut upon it of $2\frac{1}{2}$ threads per inch; the screw terminates at the loose collar r, which is upon the same shaft and bears against the piece R. The shaft is reduced in diameter when it passes through the piece R, and upon the other side of this piece it is screwed to receive a couple of lock nuts r_1 r_1, by means of which, together with the collar r, the shaft is connected with the piece R, and hence with the connecting rod Q. This rod acts in a similar manner to the connecting rod of a steam engine, so that its ends must be free to allow it to occupy the position in which it is shown in fig. 5, and the two extreme positions in fig. 6. Attached to the bottom of the headstock is a nut T (figs. 3 and 6), through which passes the screwed portion S_1 of the shaft S; this shaft, as previously stated, is free to revolve and slide in its bearings. When the elliptical wheel P turns, the connecting rod Q causes the shaft to slide in its bearings, and as the nut is attached to it, by the screw, the nut also moves, and with it the headstock. By turning the shaft S by means of the handle on the end s_1 the position of the nut upon the screw, and therefore that of the headstock upon the bed, may be raised within certain limits. We have already stated that the headstock has a range of 1 ft. 9 in.; this is made up of the crank motion, and that due to the variation in position by means of the screw and nut. The crank

motion gives a stroke of 12 in.; in fig. 5 the disc m, which forms the crank and the connecting rod Q, are shown in these positions; the two ends of the connecting rod are marked q, q^1, q^2; q_1, $q_1{}^1$, $q_1{}^2$. Upon the raised portion of the disc m, and on the left hand of the slot m_1, there is an index s (fig. 5), by means of which the centre of the end of the connecting rod is set so as to alter the travel of the drill. This index commences at the centre line $\gamma\,\gamma$, and is parallel with the slot m_1; where it coincides with the line $\gamma\,\gamma$, it is marked 0, for then the centre of the end q of the connecting rod coincides with the centre of the disc m, which is the geometrical centre of the elliptical wheel, and there is no motion of transmutation communicated to the headstock. So that if the machine were in motion, and also the gearing for moving the headstock, there would simply be a circular hole drilled by the machine. But as the centre of the end q of the rod Q is moved outwards towards the circumference of the disc m, and fixed in such position by the bolt o, a slot is drilled, and not a circular hole; the length of the slot corresponding to the length marked on the index s (fig. 5); a pointer or a mark on the end of the connecting rod indicating on the index the length. The index is marked from 0 to 12 inches, which is the length of the slot that can be drilled without moving the headstock by the screw S_1; this distance is divided into inches, and these again into sixteenths of an inch. When a slot of a certain length, which may be from 0′ to 12″, is required, the pointer is set at the required point in the index, and the end of the connecting rod Q fixed there. If the gearing be now set in motion, the headstock will have a motion of translation along the bed corresponding to this required length. This motion secures the headstock moving alternately in one direction, and then in the opposite, until the slot is completely cut.

The manner of producing the self-acting feed of the drill will be explained presently.

343. The Table.—In the front of the bed, and in the centre of its length, is a casting U, either forming part of the bed or else cast separately and bolted to it. This piece U is provided with two grooves $t\,t$ (figs. 2 and 3), of a form best shown in figs. 1 and 6, in which are fixed bolts $t_1 \ldots t_1$ for

connecting the bracket V, which carries the table W to and with the bed (figs. 1 and 2). Upon this piece U slides a bracket V (see fig. 7 also); the bracket is held in any required position by bolts and nuts $t_1 \ldots t_1$, and it is maintained in a vertical position by means of projecting pieces that fit in the parallel parts of the grooves *t*, *t*. The bracket is raised and lowered by means of a screw X, the lower end of which is carried in a bearing *u* in the frame U, and is fixed in it by the collar u_1. A worm wheel Y (fig. 7) forms the nut for this screw; this wheel is kept from moving up witn the screw by the bearing v_1, v_1, through which and the nut the screw passes. A worm Z, fastened to the shaft Z_1, which is carried by the bearings *w w* (fig. 2) of the bracket V, gears into the worm wheel Y; one end of the shaft Z_1 is squared to receive a handle *f*, by means of which the shaft is turned and the bracket V is raised or lowered, and with it the table W. In the upper part of the bracket V is fixed a nut *x* (fig. 7), through which passes a screw x_1; this screw is attached to the table W, through the front of which it passes by means of the lock-nuts *y y* and collar y_1; the outer end is squared to receive a handle, by means of which the screw is turned, and the table moved to or from the bed. The table is attached to the bracket V by the usual lip and strip arrangement (see figs. 1 and 4.) There are five T-headed slots *z*...*z* running across the table and parallel to the bed; these are for the purpose of inserting bolts for holding or fixing the piece of material, under manipulation, to the table. By the vertical and horizontal screw motions of the table the shaft or other piece of work can be brought as near as desired to the drill, the horizontal motion fixing it definitely in one direction in which the slot is to be cut, while the other is fixed by setting the headstock by means of the screw S, having due regard to the motion derived from the disc *m*.

344. The Headstock and Feed Motion.—The headstock is really a frame consisting of a web A, front and back flanges B and C, and base D, the front part *a* (fig. 2) of which forms a lip with an angular surface, which clips a similar surface on the bed. The bottom of the front of the base also bears upon two parallel surfaces on the bed, while

the back portion bears upon the back of the bed only; this portion b (figs. 1 and 4) projects beyond the bed and forms a bearing for the strip b_1, which is fixed to the headstock by the set screws $c \ldots c$; the strip is forced against the bed by the set screws $c_1 \ldots c_1$. This headstock carries a spindle, in which is fixed the drill, and the gearing for giving motion to these. The headstock has a motion of translation along the bed of 1 ft. 9 in. by the combined motion, as already explained, of the screw S_1 and the connecting rod Q. When the machine is used for drilling, and the self-acting motion is in gear, the travel of the headstock may be any length from 0 to 12 in., the reciprocating motion of the headstock being communicated to it in the manner already explained.

345. Upon the driving shaft C is a mitre bevel wheel E of 28 teeth (see figs. 4, 7, and 8). In this wheel is fixed a sliding key which fits in a groove in the shaft. The wheel has a collar bearing against the headstock, to which it is connected by a U-shaped bracket d, bolted to the web of the frame, so that the wheel slides upon the shaft when the headstock moves; and the collar of this wheel also bears against a bush e_1, fixed in the frame (see fig. 4); this bush is a bearing for the shaft C. The mitre bevel wheel E is in gear with another similar one E_1, which is keyed upon the shaft F (fig. 4); this shaft is carried in bearings d_1 and d_1; keyed upon the other end of the shaft F is a bevel wheel G of 22 teeth in gear, with another G_1 (figs. 1, 4, 7, 8) of 29 teeth upon the drill spindle H. In the bevel wheel G_1 is fixed a sliding key e (fig. 7), fitting in a groove in the spindle, so that the spindle can slide up and down in the wheel. The wheel has a long boss f which fits the spindle; the lower end of this boss is screwed, and upon it are two lock nuts $f_1\, f_1$. This boss is slightly conical, and fits in a brass bush g, which again fits in the projecting part a_1 of the frame. This bush is bored conically to receive the boss of the bevel wheel, and is maintained in its position, in the frame, by the nuts $g_1\, g_1$; its position may be raised, by means of these, so as to compensate for wear. When the spindle revolves it also turns the bevel wheel G, and at the same time the spindle is free to move up and down in the wheel. The piece of metal h, in the front of the boss a_1, is simply a plug to fill up the

hole made in it for boring out the bearings for the shaft F, and for the purpose of fixing that shaft in position.

The spindle H (fig. 7) is bored at its lower end to receive the shank of the drill K, the drills will be referred to later on; this shank is slightly tapered for a portion of its length, the end being parallel and screwed to fit in the screw portion h_1 of the spindle. The lower part of the spindle H is of larger diameter than the upper, and it passes through the bevel wheel G_1, as before mentioned, to which it is connected by the sliding key *e* fixed in the wheel. The sliding motion of the spindle is produced by a rack and pinion movement. A hollow spindle L fits upon the portion H_1 of the drill spindle, and this spindle has cut upon its back surface a rack M which is in gear with a pinion M_1. The drill spindle is free to revolve in the hollow spindle L, which motion it receives from the bevel wheel G_1; the spindle H_1 passes through the spindle L, and its end is screwed to receive a couple of lock nuts *k k*, below which there is a washer bearing upon the spindle L; at the lower end of L, and between it and the larger diameter of the drill spindle, are two steel washers *l l*, against which the portion H of the spindle bears. The wear caused by the rubbing contact of the surfaces of the spindle H with the washers *l* and *l*, and of the upper one with the spindle L, and that between the upper surface of the spindle and the washer between it and the lock nuts, is compensated for by the lock nuts *k* and *k*, allowance being made in the upper part of the spindle H for this. The hollow spindle L is carried in a bearing l_1, which is part of the headstock frame; in front of it is fixed a pinching screw *m* which forces a small brass disc against the spindle, for the purpose of taking out any shake or back-lash that may be produced by wear. The rack pinion M_1 is fixed in a space formed in the frame behind the rack, and on each side of it is a bearing through which passes a shaft N; this shaft is caused to rotate with two motions, both of which, however, act through the worm and worm wheel O_1 and O (figs. 7 and 8), which will now be described.

346. It has already been shown that the drill spindle receives its rotary motion from the bevel wheel G_1, and its motion of translation, whether downwards or upwards, is

transmitted by the rack and pinion; the downwards motion of the spindle, and therefore of the drill, is usually described as the feed motion, and it is to this that we shall now refer. Upon the rack pinion shaft N (figs. 2, 7, 8, and 9) there is a worm wheel O which is free to turn upon the shafts; on one side of this wheel is a plate connecting the rim and boss, and on the other it is open, and its inner circumference is turned conical, into which fits a cone n (see fig. 9). This cone is keyed to the shaft N, so that by forcing it into the hollow cone of the wheel O the two are connected by the friction between their surfaces; the pressure to produce this friction is given by turning the hand wheel o, the boss of which forms a nut and fits upon the screwed end of the shaft N. The boss of the hand wheel bears against that of the cone n, and by turning the hand wheel in a right-handed direction the cone is forced into the hollow cone of the worm wheel O, which is thus connected to the shaft. By turning the wheel o in a left-handed direction, the cone is free to separate from the worm wheel, which separation is aided by a spring n_1 fixed in the cone n, with its ends pressing against the plate of the worm wheel O; when this wheel is connected to its shaft the feed motion is acting, and *vice versâ* when it is disconnected. This wheel gears with a worm O_1 (figs. 7 and 8) keyed on an inclined shaft P, carried by bearings p and p_1, the latter forming part of the base D of the frame. The shaft P receives motion from one of two sources, according to the class of work to be done.

(1.) Upon the shaft P, and in contact with the lower bearing p_1, is keyed a worm wheel Q (figs. 3, 7, 8, 9) which is in gear with a worm Q_1; this worm is fixed upon a short shaft q carried by the bearing q_1. On the other side of the bearing q_1, which is cast on the frame, there is a cone pulley R of two speeds, in which is fixed a handle r for working by hand (fig. 2). Upon the driving shaft C there is a cone pulley R_1 of two speeds, in which is fixed a sliding key, so that the pulley is free to slide upon the shaft and to rotate with it. A strap connects the two pulleys R and R_1, by this means the motion from the driving shaft is transmitted to the shaft q, then by the worm and worm wheel Q_1 and Q to the shaft P, and hence by the remaining gear to the drill spindle.

When this feed motion, which is independent of the one to be next described, is not required the strap is taken off the pulleys R and R_1. By means of the handle *r* the feed may be hand-worked if desired; in addition it may be used as a means of raising the drill spindle, but this is usually done by turning the hand wheel Y (fig. 2) on the shaft N to which it is attached.

(2.) The second method of feed is the self-acting one, and the one employed for drilling slots for cotters, etc. This feed is given simultaneously with the reversal of the transverse motion of the headstock; that is, the feed is given at the instant the end of the slot is reached, and just as the drill begins to return along the slot. Commencing at the shaft P; upon the lower end of this shaft, and below the bearing p_1, is keyed a bevel wheel T, which is in gear with another bevel wheel T_1 (fig. 9) keyed on the shaft U (see figs. 7, 8, and 300). This shaft is carried in bearing *s s*,

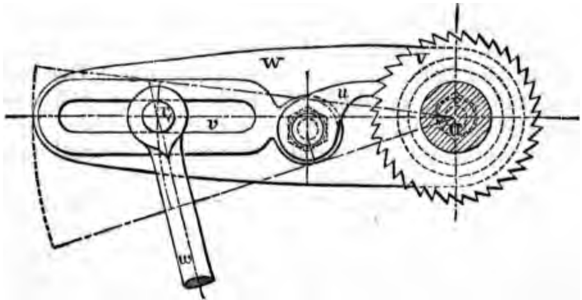


Fig. 300.

in the ends of the bed; on the inner side of one of these bearings *s* is fixed a loose collar r_1, which prevents the shaft moving outwards. The bevel wheel T_1 is carried by a bearing *t*, which is attached to the under side of the base of the headstock by screws t_1 t_1; this wheel is free to revolve in its bearings, and in it is fixed a sliding key which slides in the keyway in the shaft. Outside the bearing *s* (figs. 8 and 9) there is a ratchet wheel V keyed on the shaft U, and beyond the ratchet wheel, on the same, is a lever W, which carries the paul or catch *u* for working the ratchet wheel; this lever

is free to turn on the shaft, and is kept in position by a washer and set screw. If when the catch u is in gear with the ratchet wheel, as shown in fig. 300, the lever W is raised, the shaft is turned in its bearings, and the motion is transmitted by the bevel wheels T and T_1 to the shaft P, and hence to the drill spindle and drill. In the lever W there is a slot v, in which is fixed a pin v_1; this pin carries one end of a rod w, the other end of which is carried by a pin w_1 fixed in the slot x of the lever W_1. The lever W_1 is bell-cranked, but its arms contain an angle greater than a right angle; it is carried by a stud fixed in the bed, upon which it is free to turn, and is kept in position by the washer and nut x_1. The other end of this lever is forked, in the fork is fixed a pin y, which carries a bowl or roller X; on the under side of the disc m of the elliptical wheel P, there is a tappet, of the form shown in figs. 8 and 9. The roller X is placed in the path of this tappet, as it revolves towards and over that portion of the centre line YY, which lies outside the end of the bed at which the arrangement is fixed. In the position shown in fig. 8 the roller and tappet are in contact, and the centre of the latter is in the centre line $\gamma\,\gamma$. The centre line of the lever, to which the roller is attached, has been moved through the angle shown by the radial lines; by the time the tappet leaves the roller this centre line will occupy the extreme position shown by the outer radial line. The extent of angular motion of the end containing the slot x of the lever W_1 is also shown by radial lines, as is also that of the lever W for the position of the rod w shown. By varying the position of the rod in one or both the slots on the levers W and W_1, by moving the pins v_1 and w_1 a greater or less angular movement of the lever W can be obtained, and by this means the amount of feed increased. In fig. 300, the lever W and the ratchet wheel and catch are shown, drawn to a larger scale. When the tappet X_1 releases the roller X the centre of the roller occupies the end of the radial line, near the centre line $\beta\,\beta$, while the other end descends into a horizontal position, and with it moves the lever W. This movement of the lever causes the catch u to slip over the teeth of the ratchet and take up a new position ready for the next upright motion of the lever. When this self-acting motion

is not required, all that need be done is simply to throw the catch *u* out of gear with the ratchet wheel. The various details and kinds of motion communicated to this machine have now been described. To complete the task, figures are added to show some of the kinds of work that are done by the machine, and the forms of drills used.

347. Drills.—The drills used in slot drilling differ from those used in ordinary drilling, the latter have two angular cutting edges employed when drilling circular holes; this form of drill is unsuitable when used for slot drilling, on account of the compound motion of the drill. In figs. 301 and 302 is shown a simple forked drill, which is the one most commonly employed, and is used for "roughing out."

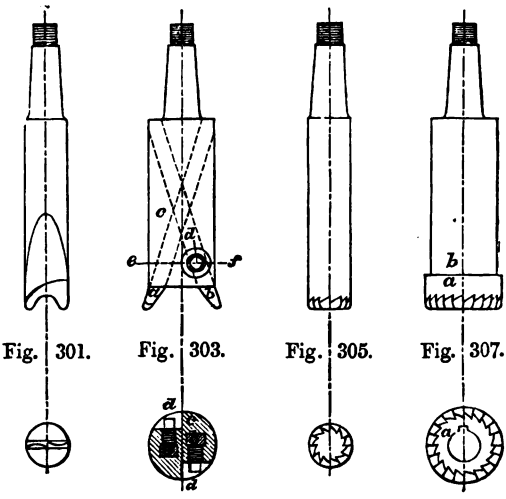


Fig. 301. Fig. 303. Fig. 305. Fig. 307.

Fig. 302. Fig. 304. Fig. 306. Fig. 308.

Fig. 302 is a plan of the cutting edges. Another form of "roughing out" drill, for large slots, is shown in figs. 303 and 304, the drill in this consists of two steel cutters *a* and *b* fixed in a socket *c*, to which they are secured by pinching screws *d d*. Fig. 304 is a sectional plan taken through *e f*. The cutters *a* and *b* can be taken out of the socket *c* for the

purpose of grinding and setting. This drill can be adjusted to cut different-sized slots. After the roughing out has been done a finishing drill may be employed if desired, two of such drills are shown in figs. 305-308. The figures 305 and 306 represent a simple rose drill for small-sized slots and narrow surfaces, the drill is in one piece, and the cutting edges are cut upon its lower surface, of which fig. 306 is a plan. In figs. 307 and 308 is shown a rose drill for finishing large slots and surfaces, surfaces of a width not greater than the diameter of the drill. This drill has a loose cutter *a* keyed to the lower end *b*. The form of the cutting, surfaces are similar in each. Fig. 308 is a plan of the cutter showing the cutting edges.

348. Specimens of Slot Drilling.—In the following figures are given a few examples of the work executed by this machine; they of course only represent a fraction of the uses to which it may be employed, but at the same time they are characteristic examples, and give a good idea to those unacquainted with the machine. The surfaces and slots cut are represented by diagonal lines in the several figures. Fig. 309 is a connecting-rod strap, in which the slots *a* and *b* have been cut, also the oil cup *c*.

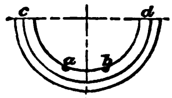


Fig. 311.

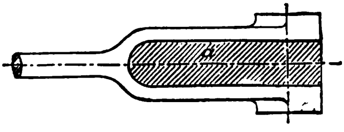


Fig. 310.

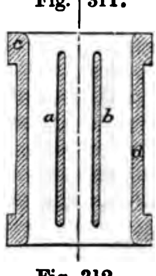


Fig. 312.

Fig. 310 is the forked end of a rod forged solid, the inner *portion a* being cut out by slot drilling.

Figs. 311 and 312 are plan and sectional elevation of a step for a pedestal, in which are cut two oil ways, *a* and *b*. The flat parts *c* and *d* are also surfaced by the drill.

349. Lathes.—A lathe in its simplest form consists of a pair of headstocks, a rest, a bed, and standards for supporting the bed. A lathe of this simple description is shown in fig. 313. The fast headstock is marked A, the loose one B,

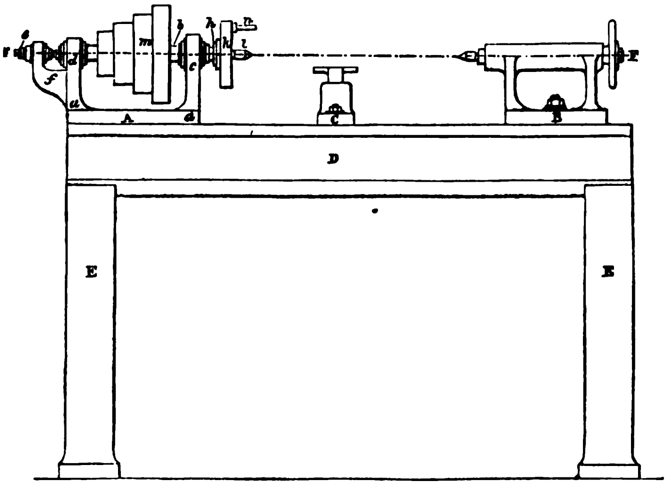


Fig. 313.

and the rest C, the bed D and the standards E E; the line F F is the centre line or common axis of the lathe; that is, the spindles of the two headstocks have this line for their common axis; this line must be level and the bed parallel with it. The loose headstock can be moved along the bed towards or from the fast headstock; the groove in the bed along which it moves and the line F F are parallel. With such an arrangement objects can be turned or bored to a cylindrical, conical, or spherical form, or a combination of these. The rest C supports the tool of the operator, which in this case is a hand tool, as the lathe is a simple single-speed hand lathe.

The fast headstock A consists of the frame *a a*, which carries the spindle or mandril *b*; this spindle is supported by bearings *c* and *d* at each end of the frame; the front bearing *c* is conical, and therefore the front neck of the spindle is conical, but the other portions are cylindrical. At the back or tail end of the spindle there is a tail pin *e*, to receive the thrust transmitted by the spindle when the lathe is in use, this tail pin is carried by a small bracket *f*, which is part of the frame A; the nuts keep the pin in position. The nose end of the spindle, that end to the right of the bearing *c*, has a collar *h*, and in front of this collar the spindle is screwed to receive the face plate *k*, this end of the spindle is bored out to receive the centre *l*.

The bearings are turned cylindrical, and fit in cylindrical apertures in the frame A, each end of the bearings is screwed and provided with a nut, for the purpose of regulating the position relatively to the frame. On the spindle *b* is keyed a cone pulley *m*, of four speeds, so that, with an ordinary countershaft, the lathe can be worked at four different speeds, depending upon the kind of material to be operated upon and the size of the object to be turned. The face plate *k* is provided with a stud *n*, the object of which is to transmit the motion of the spindle to the work under operation by means of a carrier which is attached to the work or to the mandril. The frame A, the pulley *m*, and the face plate *k*, are of cast-iron, the spindle *b* may be either of wrought-iron or steel or Bessemer steel, the tail pin *e* is of wrought-iron steeled at the end next the spindle, the stud *n* and the washers are also of wrought-iron. The centre *l* is of steel. The bearings *c* and *d* may be either cast-iron or brass. The parts B, C, D, and E need no further description, as these parts will more particularly come under discussion presently in another part of this chapter.

350. Slide and Screw Cutting Lathe.—Plates XXX., XXXI., XXXII., XXXIII. These Plates must all four be studied together in this section. Figs. 1-4 are front elevation, plan, and end elevation of a 7-inch self-acting slide and screw cutting lathe, made by Sir J. Whitworth & Co., Manchester. Figs. 5-20 are details of the same. In fig. 2 the back shaft is shown out of gear.

The lathe has the following main parts :—A fast and loose headstock, a compound disc, rest, and carriage, with bed and supporting standards; for the purpose of describing its several parts the following divisions—I., II., III., and IV. respectively—are taken, corresponding to the parts just named. The gearing and means for producing the self-acting movement will be described in division V. On the Plates will be found the Roman numbers I., II., III., and IV., referring to the divisions just made; bearing this in mind, it will be seen that on fig. 1 there is IV. on the bed, and two A's above this; on the carriage are two more A's, and still above III. The latter A's refer to the third division, the former to the fourth. Bearing this well in mind, that each division has its own letters, no confusion can happen.

351. I. **The Fast Headstock.**—Plate XXXI., fig. 5. The headstock is fixed at the extreme left-hand end of the bed; it carries the spindle and gearing for giving motion to the work that has to be operated upon.

The lathe is a double-geared one, so that there are two shafts in the headstock, the spindle A and the back shaft B (fig. 2); they are carried by the frame C (fig. 5). The spindle A is in the common centre line of the lathe, and is carried in bearings D and E; upon this spindle are the cone pulley F and spare pinion G, and a spur wheel H, all between the bearings. Outside the bearings there is the spindle nose *f*, carrying a face plate K, and attached to the face plate is a Clement's driver L; projecting from the spindle nose is the centre M; at the other end of the spindle there is a wheel N, for the purpose of transmitting motion to the gearing V, which produces, by the aid of the screw, etc., the self-acting motion of sliding, surfacing, and screw-cutting. The spindle A varies in diameter and form; the necks, supported by the bearings D and E, are conical, and both of them decrease in diameter towards the tail end; the portion *a*, between the screwed part *b* and the bearing D, is cylindrical. Outside the bearing D is a collar *e*, against which is screwed the face plate K; the nose *f* has a thread cut upon it to receive this face plate. The spindle nose *f* is bored taper to receive the centre M, and a further portion of the spindle, from the

taper-hole backwards, is bored to receive a cylindrical pin *g*, behind this pin is another pin *h*; passing through the back of the pin *h* and through the spindle, is a cotter *k*. When the centre M is required to be removed, the cotter *k* is forced sufficiently through the spindle to drive out the centre.

At *b* the spindle is screwed, and upon it are a pair of lock-nuts *l l*, in contact with the bush bearing R; these nuts receive the end-thrust of the spindle, and throw it on to the frame C. In front of the lock-nuts *l l* is a washer *m*, filling up the space between the nuts and the pinion G; outside the bearing R is a washer *n* and a pair of lock nuts *o o*, which prevent the spindle leaving its bearings and going forwards. At *p* there is keyed a wheel N, which transmits the motion to the gearing V; as this wheel is often removed, to allow for others of a different number of teeth being employed, it does not fit tightly upon the key, and therefore, to keep it in position, a washer and set screw are employed; the set screw is screwed into the tail end of the spindle. This method of keeping a wheel or pulley in position is adopted in several places in this lathe.

The bush bearings Q and R of the spindle have conical inner and outer surfaces, and they fit tightly in corresponding apertures in the frame C. Any back-lash that arises from the wear of the bearings, of the spindle and bushes, is compensated for by regulating the lock nuts *l l* and *o o*.

Upon the spindle is keyed a spur wheel H; adjoining this wheel is the cone pulley F, which has attached to it by pins (shown in fig. 5) the spur pinion G; the pulley and pinion run loose upon the spindle. The cone pulley has four speeds, F, F_1, F_2, and F_3; F_3 is solid, and forms, together with the projecting portion *q*, one bearing; the front bearing *r* is attached to a plate *s*, which fits the speed F, and is fastened to it by means of countersunk screws. Upon this plate are two projecting pieces *t t* (see figs. 16-18), each provided with a groove to receive the head of the bolt *u*, the one most convenient being used. In the rim of the spur wheel H is a groove *v*, in which is fitted the bolt *u*, this bolt can slide along the groove *v*, so as to be connected with or disconnected with the plate *s*; in fig. 17 it is shown partly in the groove of one of the pieces *t*. In figs. 16, 17, 18 the bolt is

shown in position for connecting the wheel and pulley. By mean of this bolt the wheel H and the pulley F (fig. 5) can be connected so as to drive single gear.

From the ends of the frame C, and projecting backwards, are arms $w\,w_1$ (figs. 2, 3, and 4), carrying the back shaft B, upon which is placed the spur wheel O and the pinion P; these can be thrown in and out of gear with the pinion G and the wheel H respectively. There are two grooves $x\,x$, of a semicircular cross-section, cut in this shaft to receive a pin y, which passes through the arm w and secures the shaft in one or two of the two positions, in or out of gear, according to circumstances. Both wheel and pinion, O and P, are keyed to the shaft B. When the back shaft is in gear (as shown in figs. 1 and 5) the connecting bolt u is drawn out of the groove in the plate s, and so the pulley is free to revolve upon the spindle; as the pulley revolves it transmits its motion to the pinion G, which is in gear with the wheel O on the shaft B; the motion from the pulley is therefore transmitted to the back shaft, the pinion P or this shaft being in gear with the wheel H, and the motion is transmitted to the spindle A. The face plate K is screwed on the nose end of the spindle; attached to it by means of bolts $z\,z$, fig. 19, is the Clement's driver L, in which are screwed two studs, L_1 and L_2, these studs are for the purpose of transmitting the motion to the object operated upon, or to a carrier attached to that object. An enlarged front elevation of the Clement's driver is shown in fig. 19, Plate XXXIII.

The number of the teeth, etc., in the gearing are as follows:—The pinion G has 16 teeth ·6875 inch pitch; the wheel H has 48 teeth ·6875 inch pitch; the pinion P and the wheel O are similar to G and H respectively. The form of the frame C is fully shown in the various figures; at each end there is a projecting piece C_1 (C_1), fitting the groove in the bed; connecting this piece is a web C_2 (C_2), the object of which is to strengthen that part of the frame between the bearings; the headstock is kept in position by the set screws H and K, IV, fig. 5. Nearly the whole of the headstock is in section for the purpose of showing the several parts.

352. II. The Loose Headstock—This headstock is at the right-hand end of the bed, and is seen in position in fig. 1;

but the remarks that will be made will chiefly refer to figs. 6, 7, and 20. It consists of a frame A, and spindle B (fig. 6), screw C, and hand wheel D for moving the spindle, centre E, locking arrangement F, etc. Forming part of the frame and concentric with the common centre line $\alpha\beta$ is the barrel G, which is bored to receive the spindle B. The left-hand portion of the barrel extends beyond the front portion of the frame for convenience in working the lathe. The right-hand end of the barrel is bored a little larger than the other part to receive a bush *a*, which is attached to it by set screws *b*. The spindle B is hollow, the front end *c* is bored taper to receive the centre E, the back portion *d* has a thread cut in it, forming a nut for the screw C. This screw has a left-handed square thread, of six threads per inch, cut upon it; *e* is a collar on the screw which bears against the bush *a*; the remaining portion *f* passes through the bush, outside of which is the hand wheel D, which is keyed to the portion *f* of the screw, *g* is a washer, and *h* a set screw for fixing the hand wheel. By turning this wheel the spindle is moved backwards or forwards according to circumstances. A left-handed screw is employed. By turning the handle *p* the bolt *l* can be made to force up or relieve the griping piece *q*, according to the direction in which it is turned (see fig. 20). The griping piece *q* has an inclined surface similar to that on the bolt with which it is in contact; the upper surface is cylindrical, and of the same curvature as the spindle B. The space between the end of the spindle and collar *e* of the screw is greater than that between the other end of the screw and the inner end of the centre E, so that by drawing the spindle completely in, the centre of the spindle may be found out, which it is necessary to do at times. The spindle has simply a motion of translation, and is prevented from rotating by the key *k*, which is let into the barrel, and has its head projecting into the groove in the spindle. For the purpose of preventing any motion of the spindle when the lathe is in use, a griping or locking arrangement F is employed; this is shown in fig. 20, with a section through $S_1 P_1$, fig. 6. A bolt *l*, having an inclined surface *m*, passes through the headstock, and at right angles to the centre line $\alpha\beta$; one end *n* of this bolt is screwed, and has upon it a nut *o* with a handle *p* attached to it.

The handstock can be moved along the bed to any desired position, and when in position it is fixed by means of the bolt arrangement H, fig. 6; at each end of the bottom of the frame is a projecting piece K, which fits the groove in the bed. Fig. 7 is a section through S P, showing the barrel spindle, and groove in it for the key *k* and the screw.

353. III. **The Carriage and Slide Rest.**—The following description refers chiefly to Plate XXXII., with a few references to XXX. and XXXIII. The carriage consists of the saddle A and C, figs. 1-4, which supports the slide rest. This saddle marked III rests on the bed, see fig. 9, and is connected with it by the strips B and B, which are attached to the saddle by set screws. Above this is the top part E of the slide which carries the nut D; attached by set screws to the slide E is the top slide, or simple slide rest; the compound slide rest consists of the slides C, E, F, G, the lower portion of which is marked F and the top G, and the screw for moving G is H. On the right of the screw D is a shaft K (figs. 10 and 12), upon which there is a bevel pinion L, and a spur wheel M; the wheel M gears into a smaller spur wheel N, fixed on the end of the screw D. Upon a vertical shaft O (fig. 12) there is a bevel wheel P in gear with the pinion L; on the lower end of this shaft is fixed a worm wheel T, gearing into the main screw S of the lathe. On the left-hand side of the saddle is the nut box P, and the motion for throwing the nut R in and out of gear, see fig. 13; this motion is actuated by the handle Q. The worm wheel T is in gear with the screw S, and when the screw revolves the wheel also revolves and turns the bevel wheel P, which is upon the same shaft as the worm wheel. The wheel P drives the pinion L, which in turn drives M and then N, with which it is in gear; the wheel N being fixed upon the screw D causes it to revolve, and so moves the slide E, by means of the nut attached to it. When this transverse motion is not required, the wheel N is thrown out of gear by sliding it upon its shaft into the position N_1.

When the longitudinal or sliding motion is required, the nut R is thrown into gear with the screw S, and the carriage, etc., is made to travel along the bed, either right or left, according to the direction in which the spindle is rotating.

By throwing the wheel N out of gear, and putting the handle *d* or another handle upon the shaft K at *k*, the shaft K can be made to rotate by hand and turn the bevel pinion L, and so transmit the motion to the worm wheel T, which may now act as a pinion upon the screw S, in a similar manner as a pinion acts on a rack, and thus the carriage may be moved along the bed independently of the self-acting motion. The bottom slide E (see fig. 8) of the compound slide rest may also be moved along the slide C by turning the handle *d*, but the wheel N must be first thrown out of gear.

354. The nut and nut box is shown more clearly in figs. 12-15. Fig. 12 is partly in section, *but the parts in section are not made by same plane.* There are three planes employed; one passing through the bed to expose the screw, and one through the slides C and E, and another through the boss that carries the worm wheel shaft. Fig. 13 shows the bed and screw in section. Fig. 14 exhibits the screw-nut and pins that carry the nut in section. Fig. 15 is a plan of the top piece U, and the top half of the nut. Projecting downwards from the saddle A are two pins P_1 and P_1, which form the framing of the nut box; in this box are pieces U and U, the ends of which, *u u* (fig. 14), carry the nut R, which is in two pieces. The pieces U U are connected to the nut box by pins *p* and *p*. The nut is thrown in and out of gear with the screw by moving the handle Q; in the figures the nut is out of gear; if now the handle Q is pulled towards the front of the lathe, that is, outwards, the lever W is turned through an arc of a circle, and as the lever is keyed to its shaft *w*, the shaft is also turned round; keyed to this shaft, and working in the groves *v* and *v* (fig. 14) of the pieces U U, is a plate X, in which are two circular slots *x x*; these slots are not concentric with the shaft *w*; in each slot is a pin *y*, the end of which is fixed in the piece U. In fig. 14 the pins *y* and *y* are at the ends of the slots *x x* farthest from the centre of the shaft *w*. It is therefore clear that when the plate is turned in the direction indicated by the arrows, and the pins occupy the other ends of the slots, they will be nearer to the centre *w*, and therefore R and R will close upon S, or come nearer together. The whole is so adjusted that the change of position of the pins *y* and *y* will bring the

nut into gear with the screw. By pushing the handle Q backwards, and then turning the lever W into the position shown in figs. 13 and 14, the nut is then out of gear with the screw.

355. To return again to the compound slide rest. The slide E, figs. 8 and 9, carries the slide F, and these are connected by set screws *e e*; the bottom portion of the slide F has two circular slots *f f* in it (see fig. 11), through which pass the screws *e e*. By means of these slots, the slide can be turned round its centre *c*, the angle through which it is turned being limited only by the length of the slots *f f*. In the upper portion of the slide F there is a screw H, on one end of which there is a collar and a squared portion to receive a handle for turning the screw, and thus moving the slide G; at the other end are a pair of lock nuts. The screw H works in a nut *g* attached to the slide G; this slide is provided with the usual strip arrangement, and in the upper portion of it there are four bolts, *z*...*z*. The bolts *l*...*l*, two only being used at one time, together with the plates *m m* and the nuts *n*...*n*, hold the tool *o* in position. A slide rest, differing from this only in details, is shown in section on the plates of the boring machine.

356. IV. The Bed and Standards.—These are best seen in Plates XXX. and XXXII., to which the following remarks are almost entirely confined; and on the different Plates the bed and standards are marked IV, and there only must the references be sought. The bed consists of a hollow box-frame A, open, except at intervals, at the bottom, and with an open space or groove in the top; a cross section through S P, fig. 1, in direction W X, is shown in fig. 3, and one in direction X W in fig. 9. The standards B B are in the present case two in number, one on each end of the bed, and are of such a length as to bring the common centre $\alpha\,\beta$ of the lathe to the proper elevation for a workman of ordinary height. These standards are connected to the bed by means of bolts; a cross section of one leg is seen at *c*, fig. 3. The bed is strengthened by three bars D, of the form shown in figs. 8 and 9. The top surfaces of the bed, and also the angular surfaces, against which bear the strips B and B, are planed and surfaced to receive the carriage A carrying the slide rest. Inside the bed, and pro-

jecting through each end, is the main or regulating screw S (see fig. 6); at the right-hand end it is supported in the bearing E, and is maintained in position by the lock nuts *b* and *b*; at the left-hand end it is supported by the bearing F, fig. 5, a portion of which extends beyond the end of the bed, and is marked G. Attached to the projecting bearing G is the swing frame A, fig. 4, which carries the change wheels, etc. The fast headstock is attached to the bed by the two set screws H and K, as shown in fig. 5 (IV). The loose headstock can be moved along the bed to any required position, and then firmly connected to it by the bolt arrangement H, fig. 6.

357. V. **The Screw and Self-acting Motion.**—We now proceed to describe the mechanism that produces the self-acting motion for sliding, surfacing, and screw cutting. The references are chiefly to fig. 5 (V, V). Repeating a little: the regulating screw S passes through the bed, and is supported by bearings E and F, which are part of the bed. The bearing E, fig. 6, terminates in a facing *a* just outside the bed, and in contact with the facing is one of the two lock nuts *b* and *b*, which keep the screw in position, and prevent it moving lengthwise. The bearing F (fig. 5) projects beyond the bed, and forms a bearing for the swing frame A (it is marked G and *c*.) The screw is enlarged where it passes through the bearing F, and at *d* there is a collar upon it; this collar bears against the end *c* of the bearing F, and also enters the central portion of the swing frame. The swing frame is maintained in any required position by the set screws *e* and *f*. These screws fit in circular grooves in the frame (see fig. 4), so that by loosening them the frame can be turned about its centre W. The swing frame has two grooves, *g* and *h*, in it (fig. 4), which are parallel to the common centre line; in each groove is fixed a stud for the purpose of carrying the change wheels. Only one stud, G, is shown in use in the figure, the other is marked H. By loosening the nuts at the back of the frame, these studs can be moved to any required position, according to the size of the wheels employed. In the figure the swing frame is shown in a position that will not allow of the back shaft to be used; if it be necessary to use this shaft for such a position of the swing frame, then the frame can be inclined to the right

instead of to the left. In fig. 5, V, to the left of collar is a washer *k*, and then a spur wheel K, keyed to the end of the screw, and in gear with a pinion L on the stud G. The wheel K and the pinion L are each connected to the shaft that carry them by means of a washer and set screw *l*, *m*, as before mentioned. Upon the stud G, and between the pinion L and the frame A, is a wheel M in gear, with the pinion N on the end *p* of the lathe spindle A; this wheel is keyed to the stud G. It has now been shown how the regulating screw S and the spindle A are connected. To these two revolving pieces, A and S, can be given any required velocity-ratio, by simply raising the wheels forming the train N, M, L, K; and as the thread of the screw S is constant, this velocity-ratio determines the rate of motion of the tool, and therefore the pitch of the screw which is to be cut.

Before describing the change wheels as arranged for cutting any particular pitch of screw, it is well to trace the screw to its connection with the nut, and also with the worm wheel, etc., for producing transverse motion. The motion of the spindle, whether it be in single or double gear, is transmitted to the screw S by the train of wheels N, M, L, and K, and from the screw (in some lathes a rack is used for the self-acting sliding motion) the motion is transmitted to the sliding, screw-cutting, and surfacing arrangements. For sliding, that is, the operation performed by the tool when it travels in a direction parallel with the centre line of the lathe, the nut is thrown into gear, and then the carriage A with the slides upon it is made to traverse the bed in either direction, according to the direction in which the screw rotates. The speed at which the carriage travels along the bed can be regulated by the change wheels N...K, and also, if necessary, by the introduction of others. The rate of travel depends on the size of the object operated upon, and also upon the kind of material, whether copper, brass, iron, steel, etc. For screw-cutting the arrangement is similar to that for sliding, only it is absolutely necessary that the velocity-ratio of the spindle and screw S shall be such as will cause the tool to cut threads of the desired pitch. For this purpose change wheels are employed. Appended is a table suitable for the lathe under discussion.

TABLE XVIII.

Change Wheels for the 5, 6, and 7 inch Slide and Screw-Cutting Lathes.

Showing the proper Wheels for Cutting Screws of various pitches.

Threads in 1 inch.	Wheel on Mandril.	Wheel on Guide.	Threads in 1 inch.	Wheel on Mandril.	Wheel on Stud.	Pinion.	Wheel on Guide.	Threads in 1 inch.	Wheel on Mandril.	Wheel on Stud.	Pinion.	Wheel on Guide.
1	80	20	15	20			75	51	40	85	15	90
1¼	80	25	16	20			80	52	30	65	15	90
1½	80	30	17	20			85	55	40	75	15	110
1¾	80	35	18	20			90	56	40	70	15	120
2	70	35	19	20			95	57	40	90	15	95
2¼	80	45	20	20			100	60	40	75	15	120
2½	80	50	21	80	70	15	90	64	40	80	15	120
2¾	80	55	22	20			110	65	40	75	15	130
3	80	60	24	20			120	66	40	90	15	110
3¼	80	65	25	80	75	15	100	68	25	85	15	75
3½	80	70	26	60	65	15	90	70	20	70	15	75
3¾	80	75	27	40	45	15	90	72	40	90	15	120
4	80	80	28	60	70	15	90	76	40	95	15	120
4¼	80	85	30	80	75	15	120	78	20	65	15	90
4½	40	45	32	60	80	15	90	80	30	75	15	120
5	40	50	33	80	90	15	110	84	20	70	15	90
5½	40	55	34	60	85	15	90	85	20	75	15	85
6	40	60	35	40	70	15	75	88	40	110	15	120
6½	40	65	36	40	60	15	90	90	20	75	15	90
7	40	70	38	40	60	15	95	95	20	75	15	95
8	40	80	39	40	65	15	90	96	25	75	15	120
9	40	90	40	40	75	15	80	100	20	75	15	100
10	40	100	42	40	70	15	90	104	20	65	15	120
11	40	110	44	30	55	15	90	110	20	75	15	110
12	40	120	45	40	75	15	90	112	20	70	15	120
13	20	65	48	60	90	15	120	120	20	90	15	100
14	20	70	50	40	75	15	100	130	20	75	15	130

Guide Screw Four Threads per Inch.

358. For surfacing—that is, the operation performed by the tool when it travels in a direction at right angles with the centre line of the lathe—the nut is thrown out of gear with the screw, and the pinion N put in gear with the wheel M. The screw S now causes the worm wheel T to revolve, and the motion is transmitted to the shaft K by means of

the pair of bevel wheels P and L; from the shaft K the motion is transmitted to the screw D by the wheel and pinion M and N, and thus the slide E is caused to traverse the bottom slide C, and with the slide E moves the slides F and G and the tool *o*.

359. Horizontal, Boring, and Surfacing Machine.—Plate XXXIV. On this Plate is shown a $8\frac{1}{2}$-inch centre, made by Messrs. Fairbairn, Kenway, and Naylor, Leeds. The right-hand figure is a front, and the left an end elevation; the counter shaft for driving the machine is also shown, and the whole is an example of a shade-lined finished drawing. The figures are drawn to a scale of $\frac{1}{12}$, or 1 inch to 1 foot.

Plates XXXV. and XXXVI. show on a larger scale different parts of the horizontal, boring, and surfacing machine. Figs. 3 and 4 are respectively plan and sectional elevation; the section, fig. 4, is made by a vertical plane $\alpha\,\beta$, fig. 3, passing through the centre of the spindle; the spindle B is not shown in section in this figure, but is so shown in fig. 5, where it is drawn to twice the scale of figs. 3 and 4, namely, $\frac{1}{4}$ or 3 inches to 1 foot.

Figs. 5-10, Plate XXXVI., are details of the spindle and slide rest, while figs. 11 to 28, Plate XXXVII., are details of forge or smith work for the slide rest. This Plate is an example of the manner in which forge drawings, or drawings for the smith, are sent out.

The whole machine consists of a headstock, a supporting frame on standard, a vertical sliding piece, and a compound slide rest, which is carried by the sliding piece. The machine is self-acting for boring by means of the motion of the inner spindle; for surfacing by means of the motion of the slide A; it can also be made self-acting for boring or sliding by means to be described presently. For the purpose of describing the several parts of this machine, we shall divide it into sections or divisions, as in the case of the lathe.

The following divisions are those that will be taken:—

I. The Headstock.
II. The Standard and Vertical Slide.
III. The Slide Rest and Slide carrying it, which together form a Compound Slide Rest.
IV. The Gearing for providing the Self-acting Surfacing Motion.

360. I. The Headstock.—Plate XXXV. The headstock stands at the upper left-hand corner of fig. 4, marked I, and consists of a frame A, together with the spindle gearing, etc. This frame carries a hollow spindle B, which is supported in bearings C and D on the frame; between the bearings, and on the spindle, are a spur wheel E, a cone pulley F, a spur pinion G, a washer and lock nuts. Outside of the front bearing C is the spindle nose, upon which is keyed a four-jawed chuck H. Outside of the back bearing D is a spur wheel K and a pair of lock nuts. Turning to fig. 5 we have a section of spindle B; this spindle is bored out to receive the boring spindle L, the back portion of this spindle is also bored out to receive the screw M. Upon the screw M at u, outside the lock nuts o, o, is a bush N keyed to the screw, upon which is keyed the spur wheel O. This wheel O is in gear with a wheel P, fig. 3, which is keyed upon the stud Q; this stud is carried by the arm R, which is part of the frame A. Upon the stud Q, and between the wheel P and the arm R, is a spur wheel S, gearing into the wheel K on the spindle B. Below the spindle B, and in the same centre line in plan as it, is a stud T carrying a spur wheel, fig. 4, which is in gear with the wheel K, on the spindle B, and a cone pulley V. Behind the spindle B is a back shaft W, carried in the bearings Z, Z_1 (attached to the frame), on which is keyed a spur pinion X, gearing into the wheel E, and a spur wheel Y gearing into the pinion G; we have thus a double-geared headstock similar to that of the lathe previously described. The frame A of the headstock rests upon the standard B (II), and is connected to it by set screws A_1 and A_2; at each end of the handstock are projecting pieces a and a_1, which fit in a groove in the top of the standard. Connecting these projecting pieces a and a_1, and in the centre of the frame, is a web b, the object of which is to strengthen the frame. The ends of the frame connecting the bearings receive cylindrical bushes c and d, which form the bearing surfaces for the spindle B; the inner surface of the bush c is conical, as is also the neck c_1 of the spindle. In front of the neck c is a collar e, and the nose e_1 upon which is keyed the four-jawed chuck H, this chuck is shown in section, in the elevation, and in plan below; it is

also well seen in Plate XXXIV., but the remarks are best followed in Plate XXXV. The chuck consists of a plate *k f* and a boss *g*, the plate has four rectangular slots, f_1, or grooves at right angles to each other (they are best seen in the left-hand figure on Plate XXXIV.). In each of these slots is a movable piece h_1, concentric with each groove, and passing from the periphery of the plate to the boss *g* is a screw *k*, which passes through the piece h_1. The piece h_1 is cut with a thread in which the screw *h* works, so that when the screw is turned round, the piece h_1 can be moved up or down the slot in the plate *f*. Each of the four pieces or jaws are similar, and is provided with three griping surfaces k, k_1, k_2, the object to be operated upon is fixed to the chuck by means of the jaws, and as the clutch is attached to the spindle B when it moves, the object turns with it. Each jaw is kept in position in its slot by the screw *h* and corresponding screws, and the nut and washer arrangement at the back. Each screw *h* has its outer end squared to receive a handle, by which it is turned round and the jaw moved. The wheel E is keyed to the spindle B. Adjoining the wheel is a cone pulley F of four speeds, F, F_1, F_2, and F_3, and attached to it is the pinion G; the cone pulley and the pinion are loose upon the spindle, and can revolve without turning it. The arrangement for connecting the spur wheel E, and the cone pulley F, is shown at *l*, which is similar to that of the lathe. Between the pinion G and the bearing D are a pair of lock nuts *m m* screwed upon the spindle B, and next to these is a washer *n*, which forms a collar for the spindle. On the other side of the bearing D is the spur wheel K and a pair of lock nuts *o* and *o*, also screwed upon the spindle; these lock nuts cause the boss of the wheel K to bear against the facings of the bearing D, and thus form a collar, so that the spindle B is kept in position by the lock nuts *m m* and *o o* acting upon the intermediate pinion, pieces *n* and K, and the bearing D. Inside the spindle B, fig. 5, is the boring spindle L; at *p* it is increased in size to receive the boring tool, which is fixed in the hole p_1; at *q* the spindle is bored out to receive the screw M, and at *r* there is a bush fitting into the end. The bush *r* has a collar upon it which bears against the end of the spindle, and also fits the inner

surface of the spindle B; this bush has a thread cut in it to receive the screw L, so that when the screw is turned round, the spindle can be run forwards or backwards as required. At *s* on this screw is a collar bearing against the bush *t*, which is fixed in the spindle B; and beyond this is the bush N, which is keyed to the portion *u* of the screw; the portion of the screw next to this bush is squared, to receive a handle by means of which the screw can be turned round, and the spindle L moved backwards or forwards. The wheel O will slide along the bush N, to which it is attached by the key *w*, when so moved it is thrown out of gear with P. A washer *x*, keyed to the bush, prevents the wheel from sliding off the bush. The wheel K is fixed to the spindle B by a key. The back shaft W is carried in bearings Z and Z, upon the shaft is a spur pinion X gearing into the wheel E upon the spindle B, and a spur wheel Y gearing into the pinion G; these wheels and pinions can be thrown out of gear in a similar manner to those of the lathe previously described, by reversing the pin *y* which passes through the bearing Z and fits in the groove z_1; the wheels are kept in or out of gear by fixing the pin *z* or z_1 respectively. The wheel Y and the pinion X are keyed to the shaft W. Below the spindle B is a stud T fixed in the frame A, and upon this stud are a spur wheel U gearing into the wheel K, and a cone pulley V of four speeds, V, V_1, V_2, V_3; the wheel U is keyed to the pulley V, and both are free to turn on the stud T. The pulley and wheel are kept in position on the stud by the washer and set screw t_1.

361. Having now described in detail the several parts of the headstock, the movements of the different parts will now be given. Motion is transmitted from the counter shaft (see Plate XXXIV.) by means of a strap to the cone pulley F. If the back shaft is in gear then the locking bolt is out of gear, and connection between the wheel E and the pulley F is broken (see fig. 3), and the pulley turns the pinion G, which transmits its motion to Y, which, being on the shaft W, causes X to rotate, and wheel E, which is keyed to spindle B, rotates, and with it the spindle B. The chuck H also rotates with the spindle, and any article that is attached to it. The chuck is used for holding cranks, etc., that have to

be slided, turned, or bored. The wheels E and Y have each 45 teeth $\frac{7}{8}$ inch pitch, and the pinions G and X have each 16 teeth $\frac{7}{8}$ inch pitch. The gearing of the headstock produces the following change in the velocity-ratio :—The pinion G having 16 teeth, and Y, into which it gears, also having 45; therefore the velocity-ratio of the shaft upon which G is,

$$\text{is } \frac{G}{W}=\frac{45}{16} \ldots\ldots\ldots\ldots\ldots\ldots \text{(1)}$$

From the peculiar arrangement under notice spindle C is not referred to here, but will be presently; again, the pinion G is not keyed to it. The pinion X is on shaft W and wheel E on spindle B; therefore

$$\text{Velocity-ratio } \frac{W}{B}=\frac{45}{16} \ldots\ldots\ldots\ldots\ldots\ldots \text{(2)}$$

$$\therefore \text{ Velocity-ratio } \frac{G}{B}=\frac{G}{W}\times\frac{W}{B}=\frac{45}{16}\times\frac{45}{16}=\frac{7\cdot91}{1};$$

that is, the pinion G and cone pulley F make 7·9 revolutions for one of the spindle.

The diameters of the cone pulley F are respectively $5\frac{1}{2}$, $7\frac{1}{2}$, $9\frac{1}{2}$, and $11\frac{1}{2}$ inches, and those on the counter shaft are of the same size. The fast and loose pulleys on the counter shaft (Plate XXXIV.) are each 12 inches in diameter. The spindle B can therefore be driven at four different speeds by changing the position of the driving strap connecting the cone pulleys, and an additional speed for each single speed is introduced by means of the back shaft and the gearing on the spindle, making in all *eight* speeds. If the counter shaft make 100 revolutions per minute, then, with the pulley F driving the spindle, we can have it making $100\times\frac{5\frac{1}{2}}{11\frac{1}{2}}$, or $100\times\frac{7\frac{1}{2}}{9\frac{1}{2}}$, or $100\times\frac{9\frac{1}{2}}{7}$, or $100\times\frac{11\frac{1}{2}}{5\frac{1}{2}}$ revolutions, according as the strap is on F, F_1, F_2, or F_3; that is, 47·8, 78·94, 126·6, or 209 revolutions respectively; or, speaking roughly, the spindle can be made to revolve at the rate of 50, 80, 125, or 200 times per minute; if the pulley were 6, 8, 10, and 12 inches respectively, then the spindle would exactly revolve at these rates. Let the student now suppose the back shaft in gear, and find the velocity-ratio as was done in last para-

graph. It is a good exercise to trace this with each speed pulley.

The wheels K, S, and U, have each 42 teeth, 6 teeth per inch of diameter (diametral pitch), so that the pitch circle of each is 7 inches in diameter. The wheel K being keyed to the spindle B, transmits its motion to S and U, and U being keyed to the cone pulley V, U and V revolve together; and as the wheels K and U are equal to the cone pulley, V makes the same number of revolutions as the spindle B. The motion of the pulley V is transmitted to the cone pulley, which transmits it to the self-acting motion for surfacing. The wheel R also gears into the wheel S, which is keyed to the stud Q, and its motion passes to the wheel P and from P to O; the wheel O is connected to the screw M, which has 4 threads per inch and is right handed (see fig. 5), so that when O is in gear with P, and the spindle B is revolving, its motion is transmitted to the screw, which causes the boring spindle L to move. The rate of motion of the spindle L depends upon that of B and upon the velocity-ratio of the wheels P and O. The wheels K and S being equal, therefore the stud Q rotates at the same velocity as the spindle B, the wheels O and P can be changed to produce different velocity-ratios, according to the size of the hole to be bored and the kind of material operated upon. O and P are change wheels, and this machine is supplied with the following set. The first column gives the number of cuts per inch in length of the hole to be bored for the different wheels employed.

TABLE XIX.

CHANGE WHEELS AND NUMBER OF CUTS PER INCH.

Cuts per inch.	Change Wheels.		Pitch.
	P, O.	O, P.	
30	39	45	6 teeth per inch.
44	44	40	6 ,,
58	54	58	8 ,,
86	41	43	6 ,,

Upon the stud Q, and keyed to it, is a disc v (see fig. 3), with a T-headed groove in it, in which fits a bolt v_1, carrying a piece v_2, with an eyelet in it; the bolt can be fixed in any required position in the slot. If the centre line of the bolt is in that of the disc, as shown in the drawing, it will simply revolve with the disc, and carry v_2 with it, without causing any new motion; but if the centre of the bolt and that of the disc are eccentric, then the bolt and v_2 attached to it will have an eccentric motion, and the extent of this motion can be regulated by moving the bolt v_1. The motion is employed to produce, with the aid of a rocking shaft and ratchet motion, a self-acting motion for moving the slides N, and, if necessary, A (III).

362. Materials of different Parts.—The frame A is of cast-iron, so also are the gearing, cone pulleys, plate of chuck, and bush *t*. The spindle B, the screws, stud, jaws of chuck, bush N, set screws, and plate *f*, are of wrought-iron. The bush *c* and *d* for the spindle, and *r* for the screw M, are brass. The boring spindle L is of steel.

363. II. The Standard and Vertical Slide.—(See parts marked II, Plate XXXV.). The standard B consists of a hollow box frame enlarged at its base and provided with bolt holes *a*...*a*, through which pass the foundation bolts; the base projects on the right-hand side and terminates in a boss B_1, which receives the end *e* of the elevating screw E (see bottom right-hand side of fig. 4). Upon the upper surface of the standard rests the headstock A (I), which is firmly connected to it by the set screws A_1 and A_2. On the right-hand side are projecting pieces C and C_1 (fig. 3), which form the bearing surfaces for the vertical slide D. This slide can be moved vertically up and down the frame B, by means of the elevating screw E. On the upper portion of this slide there is a cross slide F, which can be moved along it by means of the screw G and nut H (III). On the left-hand side of the standard B there is a bracket M with a bearing M_1 carrying one end of the shaft; and to the left still of the bearing is a cone pulley L, similar to V; N is the other bearing for this shaft. The vertical slide D is maintained in position by the angular projecting bearing piece *b*, which is in contact with the piece C_1 (see fig. 3) of the frame, and by the

strap *c*, which is attached to the slide by the set screws *d*...*d*. The set screws *e*...*e* are for keeping the strip in contact with the angular bearing surface of C. The screw G (lower part of III) is carried in bearings $f f_1$; at *g* there is a collar upon it which projects a little beyond the surface of the slide; against this collar bears a plate *h*, attached to the slide by set screws *k k* (see fig. 3); the object of this plate is to prevent the screw leaving its bearings when turned round; beyond the plate *h* the screw end is squared to receive a handle for turning the screw. G is a right-handed square-threaded screw, of four threads to the inch.

The under portion of the slide D is provided with bearing surfaces to receive the cross slide F, which is attached to it by the usual arrangement of an angular lip *m* on one side and a strip *n* on the other. The nut H is attached to the cross slide F; by means of the screw G and nut H the slide F can be moved towards and from the headstock, parallel with the centre line of the spindle.

In the cross slide F there is a screw O, carried in bearings *o* and o_1; this screw passes through the nut P attached to the slide A; by turning the screw the slide A can be moved in a direction at right angles with the centre line of the spindles, that is, parallel with the plane surface of the chuck H. We can thus give to the slide rest, which is carried by the slide A, two motions in the same plane at right angles to each other, and of course this motion may be either to or from any particular fixed position; and by the motion to be described presently the whole can be raised or lowered vertically. The motion at right angles to the spindle is required for surfacing, and this can be made either self-acting or workable by hand as required. Upon the end, and in contact with the collar *p*, there is a spur pinion Q in gear with a wheel R (see IV, fig. 3); the pinion Q is attached to the screw O by a key fixed in it, so that the pinion can be slided along it and then out of gear with R; this is required to be done when the slide A is to be moved by hand. Beyond the pinion Q is a washer keyed to the same, to prevent the pinion leaving it, and next to this the screw is squared to receive a handle; the end beyond the bearing o_1 is also squared. The screw is maintained in its bearings

by the collar *p*, and a washer *q*, and lock nuts *r r*. The spur wheel R is keyed to one end of a short shaft, which is carried by a bearing S attached to the cross slide; at the other end of this shaft there is a mitre-bevel wheel T in gear with one marked T_1. This gearing will be fully explained in division IV.

The vertical slide D is raised and lowered by means of the screw E and nut E_1, the end *e* of this same fits into a cylindrical hole in the projecting portion B_1 of the base of the standard; bearing upon the facing of the boss B_1 are the three steel washers *s s* and *s*, and upon the upper one bears the collar of the screw e_1, in which are four holes at right angles to each other. When the slide is to be raised or lowered, the screw E is turned round by means of a lever inserted into one of the holes. The nut E_1 fits tightly in the boss D_1 of the slide D, and is prevented from turning in it by the key *t*; upon the nut there is a collar which prevents it moving upwards, as probably it would, owing to the weight upon it, if the collar was not there. The screw E is right-handed, and four threads per inch.

364. III. **The Slide Rest.**—Figs. 6-10, Plate XXXVI., also Plate XXXIV. The slide rest is shown in detail in Plate XXXVI., drawn to a scale of $\frac{1}{4}$ or 3″ to 1′. In Plate XXXVI. is shown in elevations and plans the slide rest; fig. 6 is a plan, and fig. 7 a sectional front-elevation or longitudinal section through plane $\alpha\,\alpha$, fig. 6. Fig. 8 is a sectional end-elevation, taken in the direction Y, fig. 7, through the plane $\beta\,\beta$, fig. 6; fig. 9 is a plan of part of the bottom piece D; and fig. 8 is a sectional end-elevation or cross section. The slide rest is attached to a compound slide, of which A is the upper part of the top slide; by means of this compound slide the slide rest, as a whole, can be moved in two directions in the same plane, one parallel to the axis or centre line C C of the machine, and the other at right angles to it.

The slide rest proper consists of three movable pieces D, G, and N (figs. 8 and 10); the bottom slide G can be moved backwards and forwards along the slide A at right angles to the axis CC; it is fixed in any required position by the bolts B, B. In the top part of D there is a circular T-headed slot or groove

E, in which are placed bolts F, F. The bottom portion H of the slide G is circular, and is in contact with the slide D, the two surfaces being similar, except that D has a circular groove, and H has two bolt holes; the two slides are connected by the bolts F, F. In the common vertical centre K K of the slides is fixed a pin or pivot R, round which the slide G can be turned; when the required position is determined, the two slides D and G are firmly connected by the bolts F, F. In the top portion of the slide G there is a screw L *l* (fig. 7), which can be turned round in its bearings, but is prevented from moving lengthwise, in the direction of its axis; attached to the screw is a nut M.

N is the top slide or tool rest which slides upon G; the two slides G and N are connected by means of the inclined surface O and the strip P; the nut M is fixed to the slide N, and the motion of N is obtained by turning the screw L *l*. Attached to the slide N are four bolts and nuts, the former are marked Q; S, S are clamps by means of which, together with the bolts Q, the tool or cutter is fixed to the rest N. The strip arrangement is similar to that explained in connection with the lathe; the angle which the inclined surface makes with the horizontal plane varies between 50° and 60°, according to different makers. The screw L *l* has a circular collar T, which fits into a recess in the slide G; outside the collar is a plate U, through which passes the end *l* of the screw; a portion of the end *l* is of a square cross section, and upon this is placed a handle or lever when the screw is to be turned round. The collar of the screw and the plate are in contact; the latter is attached to the slide by means of two set-screws V, V, and thus the screw is prevented from moving lengthwise.

The dotted lines $a\,a'b'$, $c\,c'd'$, fig. 7, Plate XXXVI., show the extreme positions of the top slide N; it is advisable in most cases to show the extreme positions of moving pieces, so as to see at a glance whether or not the moving piece can occupy the positions which it is intended it should.

In addition to the scale of the drawing being given, the dimensions, of at least the principal parts, should be marked upon the drawings, even in the case of full-sized drawings. In Plates XXXVI. and XXXVII. the principal dimension

lines are shown, but the dimensions are omitted. In fig. 7, Plate XXXVI., the dimension lines are not shown, on account of the colouring, but they should be shown in the drawing.

In Plate XXXVII. are shown the pieces of the slide rest which are made by the smith; such drawings are called *Forge drawings*, drawings of *Forge work*, or *Smith work*. Forge drawings are generally made full-size, except in the case of very large pieces, and have all the dimensions added; not only those which the smith requires, but also those necessary to finish the article, as the forge drawings pass into other hands besides the smith's. The dimensions put on forge drawings are *finished dimensions*, so that the smith must make allowance for the material which has to be cut away in the different operations each piece has to undergo. It is usual to mark, in writing or otherwise, those pieces which are to be finished, as, *finished all over*, or *bright;* those not so marked being left in the *black*, that is, as they leave the smith. Two ways of marking the pieces are shown, and the quantity of each piece required. In the case of screws, worms, etc., the pitch or number of threads per inch, the hand, right or left, and whether single thread or otherwise, are marked upon the drawings; sometimes the threads are drawn by one of the approximate methods given in figs. 96 and 98, Plate VIII. There are many other notes to be made upon the drawings which depend upon circumstances, but as these vary considerably, we can only indicate those more generally used. The following figures are shown:—Figs. 11 and 12 are front and end elevation of the screw L*l* for the slide G. Figs. 13 and 14 are front and end elevation of one of the screws Q for the tool clamps S S. Figs. 21 and 22 are front and end elevation of one of the nuts for the screws Q. Figs. 15 and 16 are front and end elevation of the pin or pivot R. Figs. 17 and 18 are plan and front elevation of one of the clamps S for holding down the tool or cutter. Figs. 19 and 20 are front elevation and plan of the plate U for holding the screw L *l* in position.

365. IV. The Gearing for Self-acting Surfacing Motion.—Figs. 3 and 4, Plates XXXV., the part marked IV is what is now under discussion. Repeating a little:—The

slide A, which carries the slide rest, moves along the cross slide F, and this motion is required for surfacing objects that are fixed in the clutch H; this surfacing motion can either be produced by hand or by the self-acting arrangement now to be described. If the surfacing is done by hand, then it simply means that the wheel Q, on the screw O, is thrown out of gear with the wheel R, and the screw turned by hand as required, and the slide A, with the rest and tool, moved along the slide F. If the surfacing is done by a self-acting motion, then that motion must be of such a kind as to produce the required velocity of the slide carrying the cutting tool; and the intermediate gearing, between the screw O and the point where the motion is transmitted from, must be arranged accordingly. In the first case, the motion is obtained from the spindle B (I); the cone pulley V is attached to the wheel U, which is in gear with K; U and K are equal, so that the cone pulley V makes the same number of revolutions per minute as the spindle B, but the number varies, as we have seen, according to the position of the driving strap upon the cone pulley F, and whether the back shaft is in gear or not. A strap connects the cone pulley V with the cone pulley L below it, these are equal and similar; the pulley L is keyed to the shaft K, as shown at *l*, therefore the shaft K has the same rate of motion as the spindle B. Upon the shaft K is keyed a worm Y, gearing into a worm wheel X, which is keyed to the vertical shaft Z. The shaft Z is carried by a fixed bearing *x* and by another *w*, which is attached to the slide D, so that *w* moves up or down the shaft Z when the slide is moved. Upon the shaft Z, and carried by the bearing *w*, there is a mitre bevel wheel W_1, this wheel is attached to the shaft Z by a sliding key; the shaft is in contact with the boss of the wheel, and the boss of the wheel forms the bearing surface in contact with that of *w*. The shaft Z is of sufficient length to admit of the slide D being put in any required position. In gear with the mitre bevel wheel W_1, is another W, keyed to the horizontal shaft V; this shaft is carried by a fixed bearing *u* and a movable one U to the right of *u*. The movable one U is cast on the slide F, and carries a mitre bevel wheel T_1, the shaft V passing through the wheel, and they are con-

nected by a sliding key; the arrangement is similar to that of the wheel, shaft, and bearing, W, Z, and *w*. The wheel T_1 therefore moves along the shaft V when the slide F is moved by the screw G. A bearing S, fig. 3, cast on the slide F, carries a short shaft, upon one end of which is the mitre bevel wheel T gearing with T_1; at the other end of this shaft is the spur wheel R gearing into the pinion Q; the pinion Q is keyed to the screw O, but can be slided along its key and thrown out of gear with the wheel R. When Q is in gear with R, the motion of the shaft K is transmitted to the screw, but is not of the same velocity. This change of velocity-ratio is effected thus :—

The worm Y is $\frac{3}{8}$ inch pitch, right-handed single thread, the worm wheel has 30 teeth; therefore the velocity-ratio

$$\frac{K}{Z}=\frac{30}{1}\ldots\ldots\ldots\ldots(1).$$

The mitre bevel wheels W_1 and W have each 20 teeth, 5 teeth per inch; the shafts V and Z therefore rotate at the same speed. The mitre bevel wheels T and T_1 are of the same dimensions as W_1 and W, and therefore the shaft that carries W and the spur wheel R rotate at the same speed as the shafts V and Z. The spur wheel R has 42 teeth, 6 teeth per inch, and the pinion Q has 18 teeth; the velocity-ratio of the shaft R_1, which carries R, and the screw O, which carries Q, is

$$\frac{R_1}{O}=\frac{18}{42}=\frac{3}{7}\ldots\ldots\ldots\ldots(2).$$

As equations (1) and (2) represent the only changes in velocity, we get the following:—

$$\text{Velocity-ratio } \frac{K}{O}=\frac{K}{Z}\times\frac{R_1}{O}=\frac{30}{1}\times\frac{3}{7}=\frac{90}{7}=\frac{12{\cdot}857}{1};$$

that is, for every revolution of O there are 12·857 revolutions of K. The screw O has 4 threads per inch, so that for any 4 revolutions the slide F moves 1 inch, and to do this the shaft K must make 12·857 × 4 = 51·428 revolutions.

SECTION III.

STEAM HAMMER.

366. Steam Hammer.—The steam hammer is of more recent date than one might suppose, considering the many improvements that have been made in engineering tools up to the time of its introduction. So far as the actual working steam hammer is concerned, its invention is due to Mr. Nasmyth, of the firm of Nasmyth, Gaskell, & Co., Patricroft, Manchester; he in fact designed and patented the first that was used in Great Britain. This hammer was sketched by Mr. Nasmyth in 1838, but it was not till 1842 that it was patented, and towards the latter end of that year the first one was finished. In 1784 the celebrated James Watt took out a patent for a steam hammer, but it was never brought into practical use; this was followed in 1806 by another, by Mr. W. Deverrall, which met with a similar fate. Altogether, the steam hammer was a decided improvement on the old Tilt or Helve hammer, still there was much to be done to make it the perfect tool it has become; like the inventor of the steam engine, Mr. Nasmyth could not foresee the future; for the first steam hammer was made simply to perform the heavy work that the smithy hammer of that day could not do; the first was a moderately large one, it was, in fact, what is called a 30 cwt. hammer; that is, one having a falling head or block of 30 cwt.; this head being fixed to the end of the piston. Owing to the improvements that have been introduced, a much wider field has been opened out for their use, and now there are hammers in use of from 2 cwt. to 100 tons.

The general form of the standard, or supporting frame, of the first steam hammer is followed to the present day, but owing to the immense variety of cases in which it is now employed, other forms have been introduced to meet special requirements; but the double standard, or steam hammer framing, as it is called, of the present day, is, as we have just stated, similar to the one first made nearly forty years ago. The first hammer was hand worked; that is, the valve used to

admit steam into the cylinder was worked by hand, and the kind of valve used was the ordinary steam engine slide valve, which, owing to the pressure upon the back of it, could not be worked without considerable force, and then only slowly. To remedy this, various attempts at self-acting motion were made, but with little or no success, until Mr. Wilson invented his self-acting motion, which was at once introduced, and was used for the next ten years. This arrangement then began to give place to others, and now the larger-sized hammers are chiefly hand worked, and the smaller ones may be either hand worked, self-acting, or both.

For many years after its introduction, the steam hammer was only single-acting; that is, the steam was used only to raise the hammer head, which, when sufficiently elevated, the steam was shut off, and the head allowed to fall, under the influence of gravity alone, the steam under the piston exhausting into the air; but later on, within a very recent date, it was made double-acting, so that in addition to the force due to the falling head, there was that due to the pressure of steam acting above the piston; this was a very decided improvement, as a hammer could now be employed to do the work which had previously been done by one several times its size.

There are now, and have been for some years past, a variety of steam hammers of different makers, but all of them modifications, more or less, of the original Nasmyth hammer. The most important change, perhaps, that has been made, is that by Ramsbottom in his duplex hammer, where there are two hammer heads acting horizontally and towards each other, the metal being placed between them, and thus the anvil block proper is dispensed with.

As the space at our disposal is but limited, we have selected only one for exemplification, and that manufactured by the company representing the originators, Nasmyth, Wilson, & Co., of Patricroft; this hammer is a 15 cwt. double and single acting one, with double standards; it is hand worked, and supplied with Wilson's patent balanced slide valve and head gear. It is shown in plan and elevation in Plate XLV.

367. Steam Hammer—Wilson's Patent.—In figs. 1-5

Plate XLV., is shown in plan and elevation the 15 cwt. single and double acting steam hammer mentioned at the end of the last paragraph, and on figs. 6-15, Plates XLVI. and XLVII., are shown details of the same. Fig. 1 is a front elevation, fig. 2 a sectional plan, the right-hand half of which is made by planes $\delta \_|^{-} \delta$, fig. 1 (there are two parallel planes used, so as to make a better section, this will occur in other places). The left-hand half of fig. 2 is partly in section, which is made by the horizontal plane $\epsilon\ \epsilon$, the hammer head is not shown in section. Fig. 3 is a side elevation of the cylinder, and portion of the standards; fig. 4 is a plan of the top of a standard. Fig. 5 is a section of the right-hand standard made by the horizontal plane $\eta\ \eta$. Fig 6, Plate XLVI., is a sectional elevation of the cylinder, steam port, valves, etc., made by a vertical plane passing through the centre of the cylinder, and parallel to the centre line. Fig. 7 is a plan of fig. 6, the right-hand half is a plan of the casting, etc., between the cylinder and the standards, and is partly in section, as made by the two planes $\theta^{-}|\_\theta$; the left-hand half of the figure is a plan taken from the top of the cylinder along the plane $\lambda\ \lambda$. Fig. 8 is a side elevation of fig. 6, with the steam chest cover removed and also the back of the valve box, as made by the planes $\mu\ \mu$, fig. 9. Fig. 9 is a sectional plan made by the horizontal plane $\rho\ \rho$, fig. 8.

Fig. 10, Plate XLVII., is a front elevation of the throttle valve; fig. 11 is a sectional plan of the same, made by the planes $\tau \_|^{-} \tau$, fig. 10. Fig. 12 is a sectional end elevation made by the planes $\sigma^{-}|\_\sigma$, fig. 8. Fig. 13 is an elevation of the valve.

Figs. 1-5 are drawn to a scale of $\frac{1}{16}$, or $\frac{3}{4}$ of an inch to the foot; figs. 6-9 to a scale of $\frac{1}{8}$, or $1\frac{1}{2}$ inches to the foot; and figs. 10-15 to a scale of $\frac{1}{4}$, or 3 inches to the foot.

General Description.—The hammer consists of a pair of standards, forming the supporting frame, an intermediate piece connecting the standards, and supporting the cylinders, steam chest and valves, piston and piston-rod, hammer head, anvil, and a foundation plate, upon which rests the standards. The foundation plate, together with the anvil block, are carried upon suitable foundations. For the purpose of

describing the several parts, the following divisions will be taken :—

I. The Standards, Foundation Plate, Anvil, and Hammer Head.

II. The Intermediate Piece, the Cylinder, Valve Chest, Piston and Piston-rod.

III. The Slide Valve, the Throttle Valve, and the Gearing for working them.

368. I. The Standards, etc.—Figs. 1 and 2, Plate XLV. The standards, of which there are two, A A, form the supporting frame; each consists of a centre web B of varying dimensions, with flanges on its ends at *a* and *b*. There are ribs for the purpose of strengthening these parts. At the bottom of the standards there is a plate C forming a foot, by means of which the standard is connected by bolts to the foundation plate D. This plate is provided with lugs, *c c*, between which the foot C fits; a wooden wedge *d*, and wrought-iron one *e*, being used to key the two together. The foot rests upon a packing of wood *f*, which gives a slight elasticity to the connection; the whole being joined together by bolts passing through the hole *c′*...*c′*. The foundation plate rests upon wooden beams, and these in their turn rest upon rock-beaten soil or piles, according to the nature of the ground. The standards are connected at the crown of the arch made by them by two plates, E and E, in each of which are projecting pieces *g*, *g*, which fit in corresponding grooves in the standards; the whole being connected together by bolts *h h*. The tops of the standards are connected by bolts F and F, and an intermediate piece B; this piece rests on the top where are projecting pieces K, which fit into corresponding recesses in the piece B; the whole being connected by bolts *l*...*l*. The *inner* flange G of each standard is provided with a groove *m* (fig. 5), in which works the rib or fin *n* of the hammer head H. The hammer head consists of two pieces, the main casting H and the hammer block K, which is connected to the head by the keys *o* and *p*, the latter being of wood. On each side of the head H there is a fin *n*, which fits the groove *m*, and thus the head is maintained in position. The hammer head is attached to the piston-rod by keys *q*, *q*. A sliding

bolt L is employed to hold the hammer head in an elevated position when required. This bolt is of a rectangular cross section, and slides in a suitable bearing *r* in the right-hand frame. In the figure the bolt is shown not in use; when it is required to maintain the head in place, the handle M is pulled in the direction shown by the arrow, which causes the lever N to move about its axis O, and push forward by means of the connecting rod P the bolt L, which, passing under the lower surface of the hammer head, prevents it moving downwards. Q is a facing for a nut box, and R is a facing for the lever guard W. S is a boss for the pin which forms the axis of the lever to the left. T is a boss for the pin which forms the axis of the lever Y for working the throttle valve. The section, fig. 5, is made by the horizontal plane $\eta\,\eta$, which passes through the centre of the pin U and boss W. On this pin are the two levers *s* and *t*; the former is attached to a sliding block X, which has an inclined face, against which a fin of the hammer head presses when the head rises to that height. If the hammer head rises high enough to pass the block X, then the motion of the block is transmitted by the levers to the rod Q, which has a slot in its upper end, in which works a pin attached to the lever *w*. This lever is connected with the slide valve, and when the hammer head rises as stated, the valve is moved by the lever arrangement, and steam shut off.

The anvil Y is provided with a loose block Z, which is keyed to it by keys *u* and *v*, the latter is of wood. The lower face of the anvil rests on wooden beams, fixed vertically as piles, which are carried down to a firm bearing.

369. II. The Cylinders, etc.—Figs. 6-9, Plate XLVI. The intermediate piece B, which carries the cylinder, rests upon the tops of the standards, and is connected to them by bolts *l*...*l*; and in addition to these are two larger ones, *w* and *w* (fig. 6), passing through the lower flange f_1 of the cylinder, through the piece B, and into the tops of the standards, where they are fastened by cotters *x*, *x*. The cylinder C *y* is connected to the intermediate piece by the bolts just named, and also by a number of others marked *a*...*a* (fig. 7). In figure 6 the section of the intermediate piece B is not taken through the centre lines $\kappa\,\kappa$, but just outside the web

b, as this web runs across the piece connecting the bosses for the bolts *w*, *w*, and that for the piston. This is a common way of showing such an arrangement, instead of having to put the whole surface in section. The intermediate piece B forms a cover B^1 for the lower end of the cylinder, and also forms the stuffing box A. A gland *c* is fixed in the upper portion of this stuffing box, and another one *d* in the lower portion; and between these is the packing. The gland *d* is made in halves, as it could not otherwise be fastened, with a stepped joint, as shown in figs. 6 and 8; it is maintained in position by studs and nuts. In fig. 8, the right-hand standard is removed, and also a portion of the piece B, so as to show a portion of the gland. The piston C and piston-rod D are in one piece; the former is shown in its lowest position in all the figures. The piston is simply a cylindrical piece of wrought-iron, of the section shown in fig. 6, with a groove in it, fitted with a steel ring or spring to keep it steam-tight. When these rings are used for steam engine pistons, there are frequently more than one employed for each piston. Each end of the cylinder is bell-mouthed, and in the upper end there is a cover E, connected to the flange *f* of the cylinder by the bolts *g*...*g*. The steam ports F and G are of the usual form, F leading to the bottom of the piston, G to the top; the exhaust port H is connected with a chamber *h* (see fig. 9), which extends round the cylinder to a point diametrically opposite to the port, where the exhaust pipe K is attached to it. The steam ports F and G, and the exhaust port H, are all of the same breadth horizontally, as shown in figs. 8 and 9; the port F enters the cylinder at *k*, and G at *l*, *l*, a web *m* dividing the port into two. Round the ports there is facing, which forms the bearing surface for the slide valve; and round this facing is another *o*, against which is fixed, by the studs *p*...*p*, the valve or steam chest L. In fig. 8, the cover M of the steam chest, as seen in fig. 6, is removed. The lower portion of the valve chest forms an elbow pipe N, with a flange, against which is bolted the throttle valve O, through which passes the steam to the steam chest. On the top of the valve chest there is a bracket P, which carries the stud for the levers *v* and *w*. Passing through the top of the valve chest is the valve rod Q, the

upper portion of which slides in the bracket R fixed to the top of the cylinder; this rod passes through a stuffing box *a* and gland *r*; the latter is maintained in position by the stud and nuts.

370. III. The Valves and Gear.—For the slide valve, see figs. 6, 7, and 9, Plate XLVI.; for throttle valve, see figs. 10-12, Plate XLVII.; and for the gear for working these, examine fig. 1, Plate XLV. The slide valve here employed is one of Wilson's patent balanced valves. As before mentioned, one of the first difficulties with the early steam hammer was the amount of power necessary to move the slides, because of the steam pressure on the back; the self-acting motion was invented to overcome this difficulty. But now, owing to the improvement in valves, the hand-worked gear has come into use again for large hammers. The valve proper S is covered by a U-shaped plate T, which fits on the back surface, but allows a little clearance at the sides, see fig. 314. Fig. 314 shows the back of the plate removed; the section is made by the plane $\mu\_\sqcap\_\mu$, fig. 9; the plate T is maintained vertically by the feathers $t\,t\,t^1$, an inch wide; the two former are at the bottom, and rest on the bottom of the valve chest L, and horizontally, that is, against the valve S, supported by the spring U. This spring is required to prevent shake in the valve when there is no steam in the valve chest. The valve is supported and moved by its rod Q, which has a vertical motion. The excess of area of the lower surface of the valve over the upper, on account of the valve rod Q, adds so much to reduce the effective weight of the valve, because when there is steam in the valve chest, the pressure upon the lower surface is greater than that upon the upper.

Opposite to each port, F, G, and H, and in the piece T, there is a shallow channel or false port, similar to each of the ports, and of the exact shape of each, so that whatever pressure of steam there is on the face of the valve S, from the whole or portions of the ports F, G, and H, there is an equal pressure on the other face from the ports f^1, g^1, h^1. This is the case for all positions of the valve. The holes $u \dots u$ form a communication between the two sets of ports for such positions of the valve that otherwise would prevent the valve being balanced. In the figures on Plates XLV.

and XLVI., the valve S is shown in such a position that the three ports F, G, and H are in communication with each other. In figs. 315 and 316 it is shown in the two extreme positions; these figures are sections made by the planes $a \overline{|}\ \underline{|}\ \overline{|}\ b$, fig. 314. Fig. 314 is a front view of the valve in the position shown in fig. 315. The arrows show the direction

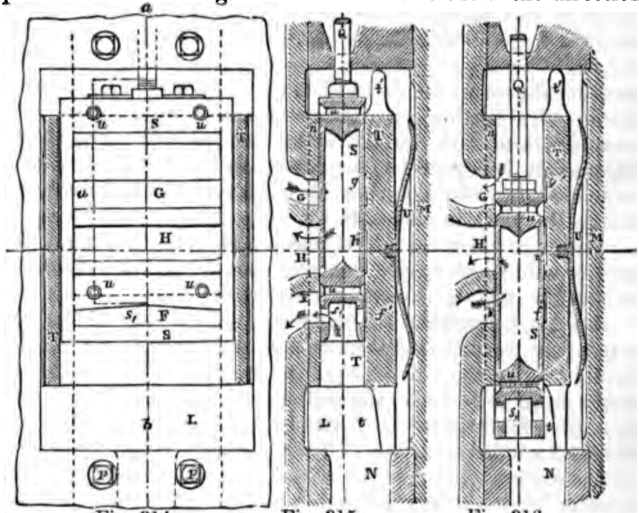


Fig. 314. Fig. 315. Fig. 316.

of the steam as it enters the lower port to move the piston up, and as it leaves the upper port for the exhaust H. The piston in this position of the slide is at the bottom of the cylinder, the valve is in its middle position, as are also the levers *v* and *w*, and the hand lever W, see fig. 1. If now the lever W be raised or turned about its axis *x*, in the direction of the arrow *y*, the connecting rod X is lowered, as is also the lever *v*, while *w* is raised, and with it the valve. The free end of the lever *v* enters a slot in the valve rod Q, so that the motion of Q may be a vertical one, which would not be the case if the end of the lever were fixed to the end of the rod Q, in which case the end of the lever would describe a small arc of a circle, and would tend to bend the rod Q in extreme

positions. The valve is raised according to the extent of motion of this lever; if it be raised into the position shown in fig. 316, then the full pressure of steam is admitted below the piston into the lower part of the cylinder, and the hammer head ascends because the port f_1 in the valve is fully open to the port F. Suppose now the piston is at the top of the cylinder, and is kept there by the pressure of steam below it; if the hammer is to be worked single-acting, then the handle W is depressed so as to bring the valve into the position shown in fig. 6, which puts the port F in communication with the exhaust port, the steam escapes, and the hammer head falls by the action of gravity. But if the hammer is being worked double-acting, then the handle is depressed further, and that in an instant, and the valve is put in the position shown in fig. 315. The steam now enters through the port G, and is free to act upon the top of the piston, while the steam that was below it runs out by the exhaust port H, as shown by the arrows.

It will be seen that the whole area of the top of the piston is free to receive the pressure of steam, so that the force acting to give the blow is equal to the pressure due to the steam upon the area of the piston, together with that due to gravity. Between the two extreme cases named, there are almost an infinite variety of others, according to the position of the valve, which can be varied at will; in addition, the force of the blow may be varied by closing the port F, and every variety of blow can be given, from a gentle tap to that already stated; for full pressure of steam, with the mass falling through the full stroke, will give a most powerful blow.

371. The Throttle Valve or Regulator O has a simple slide valve, see figs. 10 to 12, Plate XLVII. The outer casting O consists of a valve chest A, a cover B, and a cover C, which has cast to it the connecting pipe D and flange E, for connecting the whole to the elbow pipe N, fig. 6, and the valve seat *a*. The valve is a simple slide valve *b* (fig. 12), in which is fixed a nut *c*; one end of the valve spindle F enters this nut, while the other is attached to the connecting rod Y, fig. 1. Fig. 13 is a front elevation of this valve, showing also the valve seat and the facings *d d* upon which the valve

slides. Concentric with the valve spindle F there is a tube *e* forming part of the valve chest, which acts as a stop for the valve and prevents its being brought too low. A feather *f* at the top of the valve limits its upward motion. At the lower end of the valve chest there is a stuffing box G and gland H, the latter is kept in position by the studs and nuts *g g*, fig. 10. The valve chest, together with its covers and delivery pipe D, are connected by the studs and stud bolts *h*...*h*. This valve regulates the admission of steam into the valve chest of the large cylinder C *y*, the steam enters by the pipe $L K_1$, and passes in the direction indicated by the arrows through the branch pipe L into the valve chest A, then through the port *k* into the pipe D which is connected with the slide casing. The valve *b* is connected to the valve spindle F, which in its turn is attached to the connecting rod Y, fig. 1; and this rod is connected to one end of a bell-crank lever *z z*, the other end of which is connected to the rod z^1. The end of the rod z^1 opposite to that which is attached to the lever *z*, has a square-threaded screw, left-handed double thread cut upon it, and this enters a nut *z* fixed in the bracket M, see figs. 14 and 15. This bracket is fixed upon the right-hand standard as shown in fig. 1; the nut is circular and can be turned round in the boss N of the bracket, which acts as a bearing for it; on one end there is a collar *m* attached to it by pins, and at the other end a fixed collar *n*, between which and the boss N there is a hand wheel Z keyed to the nut, fig. 15. By turning this wheel the nut is turned, and the rod z^1 is moved in the direction of the arrow, or in the opposite direction, according as the wheel is turned to the right or the left; this motion is transmitted to the bell-crank lever *z z*, which in its turn causes the connecting rod Y, and with it the valve *b*, to be moved up or down, and steam admitted, shut, or cut off as required; the amount of opening regulates the amount of steam admitted into the pipe D. In figs. 10 to 12 the port *b* is closed, so that no steam can enter pipe D, or none passes to the cylinder.

SECTION IV.

STEAM ENGINES AND BOILERS.

Eight H.-P. Davy-Paxman Vertical Engines and Boiler—Robertson's Valveless Engines—Four H.-P. Horizontal Engine by Appleby.

372. The Davy-Paxman Engines, etc.—In figs. 1 and 2, Plate XXXVIII., are shown elevations of an 8 horse-power Davy-Paxman vertical engine and boiler combined; these figures show the general arrangement of the whole. Fig. 3 is a half section made by a vertical plane passing through the centre line $a\ a$, fig. 1. Fig. 4 is a half section made by a horizontal plane $\theta\ \theta$, fig. 3; on the left of this half section is shown a portion of the top of the fire-box. Details of parts of the engine and boiler are shown on Plates XXXIX., XL., and XLI.

For the purpose of describing the several parts, it will be convenient to take the annexed divisions :—

I. The Boiler and its Mountings.
II. The Bed Plate, Pump, and Feed-Water Heater.
III. The Engine.
IV. Governor and Throttle Valve.

373. I. The Boiler and its Mountings.—Figs. 1-4, Plate XXXVIII. These figures represent, in elevations and sections, the vertical boiler under discussion. The boiler consists of an outer cylindrical sheet A, fig. 3, built up of three lap-jointed tubes a, a_1, a_2, the lower edge of the bottom ring rests upon the bed plate C. The upper ring has riveted to it an end plate B, which has a circular aperture in its centre, through which passes the funnel C. Inside the shell A is the fire-box D, built up of two rings d and d_1, and a crown plate b, which is riveted to the ring d_1 and the funnel C; the lower ring of the funnel is attached to the outer shell, and rests with it upon the bed plate C. The fire-box is slightly conical, the smallest diameter being at the crown. The whole of the joints are riveted lap joints. Springing from the circumference of the lower ring d_1 of the fire-box, and at two different levels, are a number of tubes E...E, of the form shown in fig. 3. These tubes terminate a little above the crown plate b, into which they are fixed by a tube

expander; the lower ends are also fixed in a similar way. A section of one of the tubes E is shown in figs. 3 and 4, and an enlarged section of the top of a tube is shown in figs. 5-7, Plate XLI. The arrangement of the tubes in plan is shown in the sectional plan, fig. 4; on the left of the centre line of this figure, the top of the crown plate is shown, together with those of the tubes. In the crown plate, and in each tube, there is inserted a deflector F, figs. 5-7, Plate XLI.; these deflectors answer an important purpose in this class of boilers.

Referring to figs. 5-7, fig. 5 is a plan of a tube with the deflector inserted in it; fig. 6 is a sectional elevation, and fig. 7 a sectional plan made by the plan $\sigma\ \sigma$, fig. 6; the tube E projects a little beyond the top of the cover plate *b*. The deflector F has a conical body *c*, with three projecting feathers *e*...*e* which fit the inner surface of the tube; the top *f* is mushroom-shaped, as shown in fig. 6. The screwed stud *g* is used for the purpose of fixing and taking out the deflector from its tube. The tubes are cylindrical in the straight part, but in the curved conical, the smallest diameter being at *h* where they are fixed into the shell of the fire-box. The object of this tapering of the tubes is to increase the velocity of the water in its circulation through the tubes and boiler; this velocity is so great in these boilers, that if the deflectors are removed, the water would spring up like a fountain and cause priming; to obviate this troublesome evil, common in vertical boilers these patent deflectors are employed. The water rising in the tubes impinges upon the lower curved surfaces of the deflectors, which changes the direction of the flow, and causes the water to spread out over the top of the crown plate into the body of water in the boiler, and thus the mixture of the water with the steam is prevented. The tubes E...E, as shown in figs. 3 and 4, spring from the shell of the fire-box at about 1 foot above the fire-box bars G...G, and proceed towards the centre of the fire-box. They then diverge a little from this extreme position, and are inclined outwards throughout their remaining length. They are so arranged as to take up and disperse among themselves the heat arising from the fire beneath. The fire-box G...G rests upon a portion of the bed plate, and below the bars forms the ash pan A.

Another important feature in this boiler is the *baffle plate* H, which is suspended by a rod *l* at the entrance of the funnel; the heat rising up in the space formed by the inner surfaces of the tubes impinges upon this baffle plate, and is deflected by it upon the tubes E...E; thus utilising heat that would thus be lost by keeping it longer in contact with the tubes. The position of the baffle plate can be altered so as to regulate the draught; this variation in position is effected by the lever *m*, figs. 1 and 2, one end being attached to the rod *l* of the baffle plate, and the other to the rod *n*, the lower part of which is screwed, and passes through a nut in the hand wheel *o*. Hand wheel *o* is carried by a bracket K, through the boss of which passes the screwed part of the rod *n*.

Outside the outer shell A of the boiler there is a lining of felt L, and outside the whole a sheet-iron covering M; the object of the felt is to economise heat by reducing the radiation. Outside the sheet-iron covering are four hoop-iron bands *q*...*q* for binding the whole together.

374. II. Bed Plate Pumps, etc.—The bed plate B, shown at the bottoms of figs. 1, 2, and 3, is a cast-iron box frame, upon which rests the boiler and engine. It has an opening under the furnace door. The central portion A forms the ashpan and bearings for the fire-bars G. Round the ashpan the bed plate forms a tank C for the feed water which is forced into the boiler by the pump D (bottom D), figs. 1 and 2. In fig. 1 is shown a feed-water heater E connected to the pump, a section of these is shown in Plate XXXIX.; in the same Plate the heater is omitted, as the boilers are fitted both with and without the heaters. The pump is shown in section in figs. 8-11, Plate XXXIX., upon the Plate it is stated distinctly where the four sections are taken. The main casting D is connected to the tank C and bed plate E by bolts *a*...*a*; it consists of a barrel F, in which works a plunger G, and of water passages and valves. The plunger is worked by an eccentric H, fig. 1, keyed on the crank shaft D; the eccentric rod *b* is forked, and fits the end *c* of the plunger, figs. 9-11, the two are connected by a pin *d*. The upper portion of the barrel F is of larger diameter than the lower, and forms a space for the packing and

gland *e*, which is maintained in position by the studs and nuts *ff*. On the right of the plunger, fig. 9, is the water passage *g* leading from the strainer *h*, through which the water enters for the tank; in the upper portion above the strainer is a valve *k* resting on its seat *l*. This valve, like the other two *m* and *n*, is triangular in section, and has three feathers, as shown at *n*, fig. 8, but is conical where it fits upon the valve seat *l*, the top terminates in a guard or pin *o*.

In the upper portion of the valve chamber is fixed a plug *p*, against which the valve strikes each time the plunger ascends, when the pin *o* prevents it rising too high. The passage *g* is open to the barrel of the pump, and is in connection with another *q*, in the upper portion of which is fixed a valve and valve seat $m \ldots m_1$; from this valve chamber proceed two passages *r* and *s*, the former communicating with the cock *t*, through which passes the overflow, and the latter with the valve chamber *u*, by means of the valve *n*, and hence by a pipe with the boiler. The modified form of this pump, with its connected feed heater, will be noticed presently. The plugs p, p_1, p_2, are of equal diameter, and of such a size that the valve seat and valve may be inserted through the openings, which they afterwards close. The lift of the valves is the space between the top of the pins and the bottoms of p, p_1, and p_2. At the upper stroke of the plunger G the valve *k* rises in its seat, owing to decrease of pneumatic pressure in passage *g*, and water passes through the strainer into space F below the plunger; when the plunger descends it presses on the water in F, which pressure is transmitted to the top of *k*, and forces it down, and so prevents the water returning; and to the bottoms of *m* and *n*, when *m* is lifted, the water may either pass to the boiler by way of passage *u*, or, if not required, by way of *r*, to the overflow cock *t*, and so into the tank again. The cock *t* is opened by means of a key inserted through the hole t_1.

375. The Patent Feed-water Heater E is shown in section, fig. 317. It consists of a cylindrical receptacle *k*, about one-fifth of which is separated from the rest by a perforated diaphragm *w* of brass; immediately below the diaphragm is the inlet *a* for the exhaust steam, which is conveyed from the engine by the pipe L. A pipe M, provided with a

cock N, forms a connection between the space above the diaphragm and the overflow water passage *r* of the pump. Upon the top of the cylinder K, and communicating with it by the opening *y*, there is an air vessel O; the bottom of the cylinder is in communication by the opening *z* with the tank C of the bed plate. When the pump is not required for supplying feed water to the boiler, the cock *l* in the pipe L (fig. 1) is closed, and the cock N between the heater and the pump opened; the feed water then passes from the tank through the pump into the heater above the diaphragm *w*

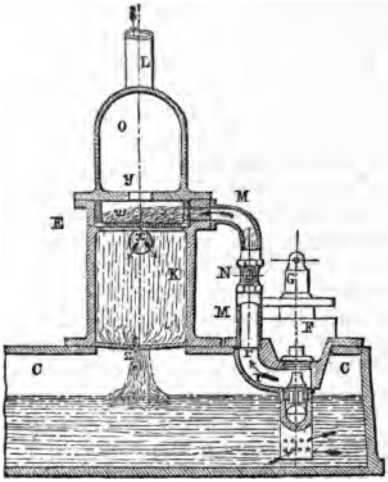


Fig. 317.

through which it falls in a fine shower, meeting the exhaust steam from the inlet *x*; the water condenses the steam, and falls to the bottom of the cylindrical receptacle, into the tank, to be again pumped up, either to the boiler or the heater, and so on. The air vessel *o* equalises the flow of the water through the diaphragm; this is necessary, as the pump is only single-acting. A portion only of the exhaust steam is used in the feed-water heater, the rest passes up the funnel to create a draught.

376. III. The Engine.—Figs. 1 and 2, Plate XXXVIII.

and Plates XL. and XLI. The engine is a non-condensing one, with inverted cylinder attached to the boiler. The general arrangement of the engine is shown in figs. 1 and 2 (III), and on Plates XL. and XLI. are details of parts. The cylinder A is bolted to the shell of the boiler by the bolts *a*...*a*; its centre line, as seen in fig. 2, is a little inclined, so as to allow clearance for the crank B and connecting rod C. The crank shaft D is supported in bearings E and E, which are carried by the brackets F, F; these brackets are connected to the shell of the boiler by bolts *b*...*b*. A stay rod G is fixed between the cylinder and the left-hand bracket; it is connected to a lug on the bottom of the cylinder by a nut, while the other end enters a boss in the bracket. The cylinder is provided with a steam chest K, in which are the valves worked by the eccentrics Y and Z; the steam chest, as well as the cylinder, is lagged, to prevent radiation. On the right, and attached to the steam chest, is the gearing V, connecting the throttle and starting valves, and above this is the governor O. The end of the piston-rod P_1, outside the cylinder, is connected to a cross head Q, the two being fastened together by a cotter. The cross head is forked to receive one end of the connecting rod C. Through the cross head and connecting rod end passes a pin, each end of which fits in blocks R; each block fits between two slide bars S S; these slide bars are suitably fastened to the cylinder cover and to the bracket T bolted to the boiler. A fly-wheel U is keyed on the crank shaft. The cylinder is 9 inches internal diameter; the stroke is 12 inches. The crank makes 125 revolutions per minute.

377. Cylinder and Slide Valves are shown in more detail in figs. 13-15, Plate XL. Fig. 13 is a sectional elevation made by the plane β, β, fig. 2, or a_1 a_1, fig. 14, which is a sectional plane made by the plane κ κ, fig. 13, while fig. 15 is a sectional plane made by the plane $\lambda\lambda$, fig. 13. The cylinder A, steam chest K, and connecting framing B, is in one casting. At each end of the cylinder is a flange *a*, upon which are fitted the covers. The top one D, and also the bottom one, are attached by the studs and nuts *b*...*b*. The lower one has a stuffing box E and piece F, to which the slide bars S′ S are attached. A gland is placed at *d*. The piston-rod,

where it fits in the piston, is conical, and is screwed to receive a nut *e*. It will be seen that the piston is formed of two cast-iron plates G and H, each with a boss; the boss *f* of the plate G has its end turned smaller than the other part, and fits a corresponding recess in boss *g* of plate H, so that there is a fixed and definite space between H and G, in which are fitted two cast-iron rings *h* and *k*, of a varying thickness, as best seen in fig. 14. The rings are cut diagonally at the smallest sections *l* and l_1, and are so arranged that if lines be drawn from *l* and l_1 to the centre of the cylinder they are at right angles. Owing to the form of these rings they act as springs, and press against the surface of the cylinder, making the piston steam-tight. The cylinder is provided with the usual ports L and M, communicating with the top and bottom of the cylinder by means of the passages L_1 and M_1, and also with an exhaust port N; this port is in connection with an outlet N_1, fig. 14, which, in its turn, is connected with the exhaust pipe L and Q, one going to the heater, the other to the funnel.

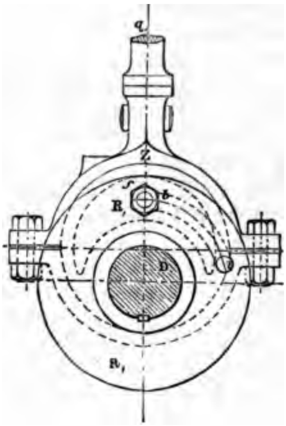


Fig. 318.

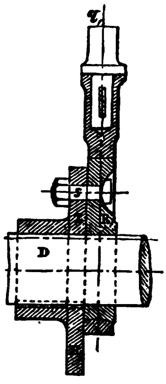


Fig. 319.

In the lower portion of the steam chest are two stuffing boxes *m* and *n*, with glands m_1 and n_1, through which the valve rods pass. It is plainly seen in the figures how the

valve rods are attached to the valves; the nut r is to regulate the position of the expansion valve Q. Both valve rods have forked ends, by which they are attached to the valve connecting rods o_1 and q_1, fig. 1, which are attached to the eccentrics Y and Z; the eccentric Y has an eccentricity of $1\frac{1}{4}$ inches, and therefore a throw of $2\frac{1}{2}''$, while Z has a throw of $3''$. The eccentric Z is loose upon the crank shaft, and is attached to a plate R_1 by a bolt and nut s; this plate is keyed to the crank shaft, and as the eccentric is attached to it, the motion of the crank shaft is communicated to it by the plate R_1. The eccentric Z moves the cut-off or expansion valve Q, according to the position of the eccentric upon the crank shaft; the cut-off is regulated by changing the position of the eccentric Z with respect to the fixed plate R_1. In the plate R_1 there are either two or more holes t_1 t, or a circular slot t t, shown in dotted lines; in one of these holes, or in the slot, fits the bolt s, and by changing the position of this bolt the eccentric Z is changed.

By the arrangement just described the steam can be cut off when from $\frac{1}{8}$ to $\frac{3}{4}$ of the stroke has been accomplished.

The valve O is an ordinary slide valve with the usual port u, and, in addition, there are the two ports v and w, which pass through the valve to its other face, upon which works the cut-off valve Q. In fig. 13 the valves O and Q are seen in their mean position, which is not the exact position for the piston shown in the same figure.

378. Connecting Rod.—This rod is shown in detail in figs. 19 and 20, Plate XL. It consists of a rod circular in cross section, and slightly tapered towards the ends A and E. The end A is attached to the crank B, and the end E to cross head Q by the pin R. The end A has two loose steps or brasses a and b, which are attached to it by the strap F, this strap being fastened to the end of the rod by the bolts c c and nuts d d. At the lower end of the strap is a boss e, through which passes a set screw f, the end of which presses against a disc g, which in turn bears against the bottom step b; by turning the set screw f the portion of the step b can be raised to allow for wear; the set screw is locked by the nut h. Within a space k is fixed the oil-cup G, in which is placed the oil for lubricating the crank bearings. The end

E is provided with brasses *l* and *m*, which are mounted in position by the strap H, which is kept in its place by a gib and cotter *n* and *o*, the cotter being driven further through the slots as the brasses wear. In this end is a slot *p*, against one end of which the cotter bears, while in the strap are two slots *q* and *q*, against which the gib bears. If the cotter be driven in the direction shown by the arrow, the step moves in the direction indicated by arrow *s*, and thus the steps are brought together.

379. IV. The Governor, Starting and Throttle Valve are shown in Plate XLI., figs. 21-23. Fig. 21 is an elevation taken in the direction of the arrow in fig. 1, Plate XXXVIII.; fig. 22 is a sectional elevation made by the plane $\gamma\ \gamma$, fig. 21; and fig. 23 is a sectional plan made by the plane $\mu\ \mu$.

The frame A of the governor is fixed upon the top of the casting V, which contains the slide valves; this casting also forms the cover for the steam chest, with which it communicates. In the lower portion E of this casting, is fixed the starting valve B, fig. 23; the valve is circular, with a passage *a* occupying nearly three-quarters of its circumference; this is in communication with the steam passage *b*, which in turn communicates with X in the cylinder casting, fig. 15, and hence by the opening *x* with the boiler. In the position shown in fig. 23, the valve B closes the opening *c*, which connects the throttle valve C, fig. 22, with the cylinder F, in which it works; the valve B is turned about its axis, and the passage *c* opened, so as to allow the steam to pass to the throttle valve, by turning the valve rod *r*, which is attached to rod *s*, fig. 1. The lower end of the rod *s* passes through the starting handle *y* into the bracket W, the top portion of which forms a plate upon which *y* works. On each side of the handle *y* there is a stop *t*, which regulates the opening and closing of the valve; the upper portion of the rod *r* passes through a gland *d* and stuffing box *e*. The steam passage *c*, fig. 22, opens into a chamber *f*, which forms the lower portion of the barrel F, in which works the throttle valve C; the chamber *f* and barrel F are in connection. The valve C has two circular plates *g* and *h*, which are connected by the spindle *k*; above each plate is a free passage for the steam, except just where the feathers *l* and *l* are (in the figure the

section plane passes through these feathers). The valve is full open in the position shown, so that the steam passes through the openings in the lower portion of the valve into the chamber F_1, through the opening D into the steam chest K; at the same time the upper portion is filled with steam, which presses upon the top of the plate *h*, so that the valve is in equilibrium, or nearly so. The valve C is suspended by the rod *m*, where it enters the upper portion of the spindle of the valve, and is fixed to it by a pin *i*; this valve rod passes through a stuffing box *n* and gland n_1 in the lower part of the governor frame A; it terminates in a collar *m* in the hollow socket *o*. In the upper part of the socket fits one end of a spindle *p*, the two being connected by a pin. Surrounding the spindle *p* and socket *o* is a hollow spindle G, the lower end of which fits in a boss H of the frames; about the middle of the spindle is a collar *q*, and above this are two grooves r_1 and r_1, through which passes a pin p_1. This pin also passes through the weight L and the spindle *p*, so that as the weight L rises or falls (it is shown in the figure at its lowest point), it raises or lowers the spindle *p*, and with it the valve C. The hollow portion of the spindle G terminates a little above the tops of the grooves r_1 r_1, it is then solid to the end, except where the slot s_1 is. The collar *q* rests upon a bush *t*, which fits in the boss M of the frame A. In the slot s_1, in the centre line $\gamma\,\gamma$, are fixed two levers N and N, centred at O, on a pin that passes through the spindle G; these levers have curved arms *u* and *u* of a parabolic form, upon which rests the dead weight L; the ends of these levers terminate in spheres or governor balls Q Q. The object of the dead weight L is to correct the variation in the centrifugal force of the governor balls. Below the boss M, and keyed to the spindle G, there is a mitre bevel wheel R, in gear with a similar one S, keyed to the spindle T. The spindle is carried by a bearing U, which forms part of the frame; a pulley *v* is attached to the outer end of this spindle. The motion of the crank shaft D is transmitted from the pulley *w*, fig. 1, by the strap w_1 to the pulley *v*, and then to the mitre wheels S and R, and as R is fixed on the spindle G, that revolves, and with it the weight L and governor balls Q and Q. and the spindle *p* and socket *o*. This governor is

arranged to make 250 revolutions per minute. Variations above or below that speed cause the spindle G and its attachment to rise or fall, and move the valve so that the steam passage F is more or less open or closed, and the amount of steam admitted to the steam chest is properly regulated. The rod *m* is made small because it does not revolve, but only acts in a lateral direction, as a rod or pillar, to open or close the throttle valve. As the engine increases its speed, the balls fly open through centrifugal force, and the arms or cams *u u* act immediately on the weight, and as the arms of the governor rise, the leverage of the weight L increases on these small cams *u u*; this weight is so arranged as to stop all "racing."

380. Robertson's Valveless Engines. — Plate XXIV. gives two views of this interesting engine, or rather donkey pump, with details of the cylinder (fig. 3) and piston (figs. 4-8), showing the peculiar arrangement by which the steam is admitted to the cylinder to drive the piston up and down, and how it passes to the exhaust. The piston itself regulates the passage of the steam to and from the cylinder, and cuts off the steam instantaneously at any point of the stroke; the exhaust passage is kept full open for nearly the entire length of the stroke.

Figs. 1 and 2 are external elevations of the pumps entire; fig. 3 is a sectional elevation of the steam cylinder, showing the steam and exhaust ports in the piston and cylinder. Fig. 4 is an external elevation of the steam piston, showing the inlet ports adapted to cut off the steam at about half-stroke. Fig. 5 is an elevation of the opposite side to that shown in fig. 4; it shows the exhaust ports in the piston, which are full open for nearly the whole length of the stroke. Fig. 6 is the piston in cross-section at the middle, while fig. 7 is an end view of the same. Fig. 8 is an end view of the crank pin driver, which is shown in section at D at the bottom of fig. 4, with the brasses and portion of the crank pin; without a careful study of this part, the student will not understand how rotation is communicated to the fly wheel. An external side elevation of this crank pin driver is seen at A in the lower middle part of fig. 1. The pump has two valve chests (inlet and outlet) B and C.

381. Appleby's 4 H.-P. Horizontal High Pressure Engine

—Plates XXV.-XXVIII.—The engine is of 4 horse-power nominal, horizontal direct-acting, and fitted with expansion gear, feed pump, and governors.

382. Steam Cylinder.—The steam cylinder has the usual three ports, is bored 6″ internal diameter, and of sufficient length for a stroke of 12″. The slide jacket is cast to the cylinder. Provision is made in the cylinder casting for flanged steam and exhaust pipes, cylinder waste cocks, lubricators, and indicator cocks, and also for carrying the lagging round the cylinder and the slide jacket. The back cover is recessed into the cylinder, the ends of the cylinder being enlarged for the purpose of reboring the cylinder when required. The cover is turned bright on the outside and recessed for the piston nut, and bolted to the cylinder by six $\frac{5}{8}''$ studs with case-hardened nuts. The front cylinder cover is cast to the bed plate of the engine. The slide jacket cover has a faced joint, while the outside of the cover is strengthened with cross ribs and bolted to the valve box by ten $\frac{1}{2}''$ studs and nuts. The stuffing boxes for the main valve and expansion valve are cast to the slide jacket, and fitted with oval gun-metal glands.

383. Piston.—The piston has a single ring $\frac{1}{2}''$ thick, divided into three parts, with brass tongue and steel spring. It is fitted on to the piston-rod with a taper and $\frac{7}{8}''$ nut. The piston-rod is of Bessemer steel $1\frac{1}{8}''$ diameter (*see* Plate XXVII.

384. Crosshead.—The crosshead is of wrought-iron, bored to receive the piston-rod, and slotted to take a brass or step for the crosshead pin; the wear of these brasses being taken up by taper cotter fitted with lock nuts on the top end. The slippers of the crosshead are of gun-metal fitted with adjustable screws and locks, as shown in the detail drawing, Plate XXVII.

385. Connecting Rod.—The connecting rod is 2 ft. 2 in. from centre to centre, tapered; the crosshead end is forked and bored to receive a 1″ diameter pin. The crank pin end is fitted with gun-metal bearing.

386. Disc and Crank Pin or Crank Plate.—The disc carrying the crank pin is of cast-iron turned bright, it is 1 ft. 4 in. diameter by 2 in. thick, and is fitted with wrought-

iron crank pin turned in the bearing, and is keyed into the crank shaft with sunk steel key.

387. Crank Shaft.—The crank shaft is of wrought-iron, 2½ in. diameter, 3 ft. from centre to centre of the bearings. The bearings are of hard gun-metal fitted into angle blocks, one being cast to the bed plate of the engine, and the other made suitable for fixing on to a stone block or into a wall box. The guides are fitted with lubricators.

388. Eccentric Shafts, Straps, Etc.—The eccentric shafts are of cast-iron, truly turned on the face and between the flanges. The main valve eccentric shafts being keyed to the crank shaft, and the expansion eccentric sheave is loose on the crank shaft, the position can be adjusted to the main eccentric by a concentric slot in it, and stud and lock nut; the two eccentric sheaves being indexed together to show the point of cut off with the valves. The eccentric straps are of gun-metal fitted with lubricators. The eccentric rods are one inch diameter at the centre. The valve rods are of steel, fitted with slotted female screw ends, and are ⅝-in. diameter where they pass through glands, and 1 in. diameter at the joint ends. This enlarged portion working in a double gun-metal guide, as shown in Plate XXVII.

389. Governors.—The governors run at a high speed, the throttle valve being formed by a piston attached direct to the spindle of the governors. The stop valve forms the casing for the throttle valve, and also the stand for the governors. The governors are driven by a light strap and bevel wheels from a pulley on the crank shaft of the engine, Plate XXV.

390. Pump.—The pump is in one casting, with the valve seats, passages, etc., formed in it. The plunger is worked from the expansion eccentric, and has a stroke of 2 in., Plate XXV.

391. Fly Wheel.—The fly wheel is 4 ft. 6 in. diameter, turned on face, edges, and boss, being keyed to the crank with a sunk steel key. The bed-plate, guides, front cylinder cover, and main block, are in one large casting, the guides being planed to receive the crosshead slippers; the bottom guide forming an oil channel; the front cover is faced to receive the cylinder, which is bolted up to it. The bed plate is one *foot* in depth from the centre of the engine by one foot one

inch wide, and lugs are provided for six $\frac{3}{4}$-in. holding-down bolts; and facings are provided for receiving the pump and the slide valve guide bracket, Plate XXV., etc.

While the student is referred to Plates XXVI.-XXVIII. for the details, Plate XXV. will give the general parts. The same letters point out the same pieces in figs. 1-3. A is the governor; B the starting gear; C *y* the steam cylinder; G the guides, between which work the slide block S B; C is the connecting rod; and D the drum or pulley to actuate the governor by means of the pulley *p* and a belt. P is a large pulley to drive any piece of mechanism, such as the double action pumps, Plate XLVIII. F is the fly wheel, and P the pumps to supply water for the boilers, etc.

392. Appleby's Pair of 6-in. Double-acting Pumps—Plate XLVIII.—The pumps have two vertical barrels A and B, each double-acting, driven from a double set of gear C, D, carried by a central column E bolted to the bed plates F, the same bed plate also carrying the pumps; the column is placed in connection with the pumps to form an air vessel for them. Each pump is 6-in. diameter, with a stroke of 12 in., double-acting, with gun-metal fittings, the guards are of gridiron shape, with india-rubber discs. The whole of the valves, guard, etc., can be removed and readily replaced, as seen at G, one cover on the valve box giving access to the whole of the four valves. The suction and delivery pipes are H and I. The connecting rods are of wrought-iron, forked at crosshead end and at the crank pin end, as seen at J. The guides K to the top of the piston-rod is formed by a casting and stays attached to the column. The wheel and pinion at C and D have teeth in the proportion of about 6 to 1, the teeth being formed by means of the odontograph; the larger wheels are employed as discs, having the crank pin fixed at the proper radius for driving the pumps. The fast and loose driving pulleys L and M are 2 ft. diameter by $4\frac{1}{2}$ wide. The column or air vessel is of cast-iron, 8 in. diameter at base, by 6 in. diameter at top, with $\frac{3}{4}$-in. thickness of metal, the top cover is formed by the pedestal of the bracket carrying the crank and pinion shaft. The whole is worked by two belts passing over the pulleys L and M.

INDEX.

COLLINS' SERIES OF SCHOOL ATLASES—Continued.

HISTORICAL GEOGRAPHY.

	s.	d.
THE POCKET ATLAS OF HISTORICAL GEOGRAPHY, 16 Maps, 6½ by 11 inches, mounted on Guards, Imperial 16mo, cloth,	1	6
THE CROWN ATLAS OF HISTORICAL GEOGRAPHY, 16 Maps, with Letterpress Description by Wm. F. Collier, LL.D., Imperial 16mo, cloth, ...	2	6
THE STUDENT'S ATLAS OF HISTORICAL GEOGRAPHY, 16 Maps, with Letterpress Description by Wm. F. Collier, LL.D., 8vo, cloth, ...	3	0

1 Roman Empire, Eastern and Western, 4th Century.
2 Europe, 6th Century, shewing Settlements of the Barbarian Tribes
3 Europe, 9th Century, shewing Empire of Charlemagne.
4 Europe, 10th Century, at the Rise of the German Empire.
5 Europe, 12th Century, at the Time of the Crusaders.
6 Europe, 16th Century, at the Eve of the Reformation.
7 Germany, 16th Century, Reformation and Thirty Years' War.
8 Europe, 17th and 18th Centuries.
9 Europe at the Peace of 1815.
10 Europe in 1870.
11 India, illustrating the Rise of the British Empire.
12 World, on Mercator's Projection, shewing Voyages of Discovery.
13 Britain under the Romans.
14 Britain under the Saxons.
15 Britain after Accession of William the Conqueror.
16 France and Belgium, illustrating British History.

CLASSICAL GEOGRAPHY.

THE POCKET ATLAS OF CLASSICAL GEOGRAPHY, 15 Maps, Imperial 16mo, 6½ by 11 inches, cloth lettered, ...	1	6
THE CROWN ATLAS OF CLASSICAL GEOGRAPHY, 15 Maps, with Descriptive Letterpress, by Leonhard Schmitz, LL.D., Imperial 16mo, cloth lettered, ...	2	6
THE STUDENT'S ATLAS OF CLASSICAL GEOGRAPHY, 15 Maps, Imperial 8vo, with Descriptive Letterpress, by Leonhard Schmitz, LL.D., cloth lettered, ...	3	0

1 Orbis Veteribus Notus.
2 Ægyptus.
3 Regnum Alexandri Magni.
4 Macedonia, Thracia, &c.
5 Imperium Romanum.
6 Græcia.
7 Italia, (Septentrionalis.)
8 Italia, (Meridionalis.)
9 Armenia, Mesopotamia, &c.
10 Asia Minor.
11 Palestine, (Temp. Christi.)
12 Gallia.
13 Hispania.
14 Germania, &c.
15 Britannia.

Historical and Classical Atlas.

THE STUDENT'S ATLAS OF HISTORICAL AND CLASSICAL GEOGRAPHY, consisting of 30 Maps as above, with Introductions on Historical Geography by W. F. Collier, LL.D., and on Classical Geography by Leonhard Schmitz, LL.D., with a Copious Index, Imperial 8vo, cloth, ...	5	0

COLLINS' SERIES OF SCHOOL ATLASES—Continued.

SCRIPTURE GEOGRAPHY.

	s.	d.
THE ATLAS OF SCRIPTURE GEOGRAPHY, 16 Maps, with Questions on each Map, Stiff Cover, ...	1	0
THE POCKET ATLAS OF SCRIPTURE GEOGRAPHY, 16 Maps, 7½ by 9 inches, mounted on Guards, Imp. 16mo, cloth, ...	1	0

1 Ancient World, shewing probable Settlements of Descendants of Noah.
2 Countries mentioned in the Scriptures.
3 Canaan in the time of the Patriarchs.
4 Journeyings of the Israelites.
5 Canaan as Divided among the Twelve Tribes.
6 The Dominions of David and Solomon.
7 Babylonia, Assyria, Media, and Susiana.
8 Palestine in the Time of Christ.
9 Modern Palestine.
10 Physical Map of Palestine.
11 Journeys of the Apostle Paul.
12 Map shewing the prevailing Religions of the World.
13 The Tabernacle in the Wilderness.
14 Plans of Solomon's and Herod's Temples.
15 Plan of Ancient Jerusalem.
16 Plan of Modern Jerusalem.

BLANK PROJECTIONS AND OUTLINES.

THE CROWN ATLAS OF BLANK PROJECTIONS, consisting of 16 Maps, Demy 4to, on Stout Drawing Paper, Stiff Wrapper, ...	0	6
THE CROWN OUTLINE ATLAS, 16 Maps, Demy 4to, Stout Drawing Paper, Stiff Wrapper, ...	0	6
THE IMPERIAL ATLAS OF BLANK PROJECTIONS, consisting of 16 Maps, Imperial 4to, on Stout Drawing Paper, Stiff Wrapper,	1	6
THE IMPERIAL OUTLINE ATLAS, 16 Maps, Imperial 4to, Stout Drawing Paper, Stiff Cover, ...	1	6

A Specimen Map of any of the foregoing Atlases free on receipt of two Penny Stamps.

SCHOOL-ROOM WALL MAPS.

Printed in Colours, and Mounted on Cloth and Rollers, Varnished.

CHART OF THE WORLD, 5 ft. 2 in. by 4 ft. 6 in., ...	20	0
CENTRAL AND SOUTHERN EUROPE, 5 ft. 2 in. by 4 ft. 6 in.,	20	0
EUROPE, ASIA, AFRICA, NORTH AMERICA, SOUTH AMERICA, ENGLAND, SCOTLAND, IRELAND, PALESTINE, INDIA, each 3 ft. by 2 ft. 5 in., ...	6	6
UNITED STATES OF AMERICA, 3 ft. 11 inches by 2 ft. 4 in.,	8	6

COUNTY WALL MAPS.

Printed in Colours, and Mounted on Cloth and Rollers, Varnished.

MIDDLESEX, LANCASHIRE, YORKSHIRE, WARWICK, DURHAM, CUMBERLAND, DERBYSHIRE, DORSET, GLOUCESTER, HAMPSHIRE, SOMERSET, STAFFORD, AND WILTS; each 54 in. by 48 in., ...	9	0

CHART OF METRIC SYSTEM.

CHART OF THE METRIC SYSTEM OF WEIGHTS AND MEASURES. Size 45 in. by [illegible]	9	0